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Influence of Unheated Starting Sections on the Heat Transfer From a Cylinder to Gas Streams Parallel to the Axis[1]

By WILLIAM TESSIN[2] AND MAX JAKOB,[3] CHICAGO, ILL.

The present investigation, sponsored by the U. S. Naval Postgraduate School and performed in Illinois Institute of Technology, is an extension of the work of Jakob and Dow dealing with the heat transfer between a cylinder and air flowing parallel to the longitudinal axis. Spherically tipped nosepieces of various lengths were attached to an electrically heated cylinder. The results show that the heat transfer is a function of the ratio of heated length to total length. Comparison of the results with those of Jakob and Dow indicates a decrease of the heat transfer with increasing cylinder radius. Extrapolation to infinite radius (flat plate), however, is uncertain. Therefore, it is recommended that an average value of the heat-transfer coefficient be used independent of the curvature pending further studies of the influence of configuration and turbulence level.

Nomenclature

The following nomenclature is used in the paper:

B_0 = empirical curvature correction factor, dimensionless
b = boundary layer thickness, ft
C_0 = constant
C = circumference of cylinder, ft
F = theoretical curvature correction factor, dimensionless
F_s = function of x/L, dimensionless
h = local convective surface coefficient, B hr^{-1} ft^{-2} F^{-1}
h_m = mean convective surface coefficient, B hr^{-1} ft^{-2} F^{-1}
k = thermal conductivity of air at film temperature t_f, B hr^{-1} ft^{-1} F^{-1}
L = total length, ft or in.
m = constant exponent
$(N_{Nu})_L = \dfrac{h_m L}{k}$ = Nusselt number based on total length, dimensionless
$(N_{Nu})_x = \dfrac{h_m x}{k}$ = Nusselt number based on thermal length, dimensionless
$(N_{Nu})_{x,\,loc} = \dfrac{hx}{k}$ = local Nusselt number, dimensionless
N_{Pr} = Prandtl number, dimensionless
$(N_{Re})_L = \dfrac{v_0 L}{\nu_0}$ = Reynolds number based on total length, dimensionless
$(N_{Re})_x = \dfrac{v_0 x}{\nu_0}$ = Reynolds number based on thermal length, dimensionless
n = constant exponent
q = heat transferred by convection, B hr^{-1}
q_d = heat loss at downstream end of specimen, B hr^{-1}
q_e = electrical heat energy released, B hr^{-1}
q_n = heat loss at nosepiece, B hr^{-1}
q_r = heat loss by radiation, B hr^{-1}
S = surface area, sq ft
s = starting length, ft or in.
$t_f = t_0 + \theta_m/2$
t_0 = jet air temperature, deg F
t_w = outer tube wall temperature, deg F
v_0 = jet velocity, ft sec^{-1}
x = thermal length, ft or in.
ν_0 = kinematic viscosity of jet air, ft^2 sec^{-1}
ϕ_1, ϕ_2, ϕ_3 = configuration functions, dimensionless
θ = local temperature difference between surface and jet air, F
θ_m = mean temperature difference between surface and jet air, F

NOTE: A place on the Fahrenheit temperature scale is indicated by deg F. Differences of temperature are indicated by F.

Introduction

Heat transfer between a free air stream flowing parallel to a cylindrical surface of relatively small curvature has been measured by Jakob and Dow (1).[4] A theoretical treatment by Latzko for a flat plate (2) seemed to yield smaller values of heat transfer. Measurements of Juerges (3) and Élias (4) and their correlation by Colburn (5) had led to higher values for the flat plate.

Jakob and Dow, in correlating their experimental results, considered the influence of unheated starting lengths using a fixed heating length. More recently, Rubesin (6) dealt with this influence theoretically.

The present investigation was undertaken to find by new experiments in what sense curvature influences the heat transfer, and to study the effect of hydrodynamic starting lengths by also varying the thermal lengths.

While Jakob and Dow had used a cylinder of 1.3 in. diam, a heating length of 8 in., and starting lengths of 0.9 to 12.3 in., approximately, in the present work a cylinder 0.624 in. diam, heating lengths from 4 to 28 in., and starting lengths from 0.5 to 24 in., approximately, were employed. The choice of a

[1] Based upon a thesis submitted by William Tessin to the Graduate School, Illinois Institute of Technology, in partial fulfillment of the requirements for the degree of Doctor of Philosophy in Mechanical Engineering.

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NOTE: Statements and opinions advanced in this paper are to be understood as individual expressions of the authors and not those of the United States Navy or of the Society. Manuscript received at ASME Headquarters, Dec. 12, 1951. Paper No. 52—F-21.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

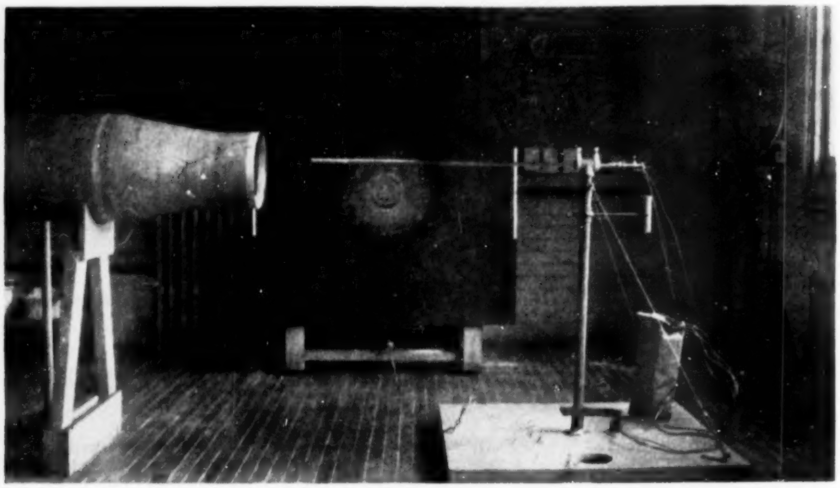

Fig. 1 General Arrangement of Apparatus

smaller diameter than that used in the previous experiments promised to show more clearly the influence of curvature.

Description of Apparatus

A blower served to provide an air stream. Dampers at the inlet in conjunction with a variable-speed drive were used to control the air velocity. A duct 14 in. diam and 29 ft long ended in a nozzle $9^{7}/_{8}$ in. diam which is seen in Fig. 1. The figure further shows the heating element, located in an air stream and clamped to a stanchion which is mounted upon a wood base. Rubber shock mountings between the base and laboratory floor damped vibrations. An additional dynamic vibration absorber was attached to the stanchion. Pieces in the background belong to another investigation.

Details of the heating element are shown in Figs. 2 and 3. The main parts were an outer tube, an inner tube, and a heating unit. The outer tube was of brass, 0.624 in. OD, 0.384 in. ID, and 35 in. long. The wall of the outer tube had three longitudinal holes, equally spaced, $^{1}/_{16}$ in. square for the insertion of thermocouples. The outer surface was bright chromium plated.

The inner tube served to locate the heating unit centrally and to provide a means for the attachment of wooden nosepieces. This brass tube was 0.376 in. OD and 0.326 in. ID.

A Calrod heating unit, 0.316 in. OD, was modified to provide a heater beginning at its upstream end, the electrical terminals being placed at the opposite end.

Cylindrical nosepieces with hemispherical tips were made of maple dowel stock.

The thermocouples were of copper and constantan, Number 30 B and S gage. Three movable thermocouples were provided for insertion into the holes of the heating element. They were connected to a Leeds and Northrup portable precision potentiometer through a rotary selector switch. Four thermocouples connected together in series, were located in the air stream at the nozzle mouth and were connected to a Leeds and Northrup Speedomax indicator-recorder which could be checked with the portable precision potentiometer. We attached a vernier to the pointer of the Speedomax so that readings to 0.01 millivolt could be made.

An impact tube in the duct, 3 ft upstream from the nozzle, was used to determine the air velocity.

A motor generator set with automatic voltage control supplied d-c heating current.

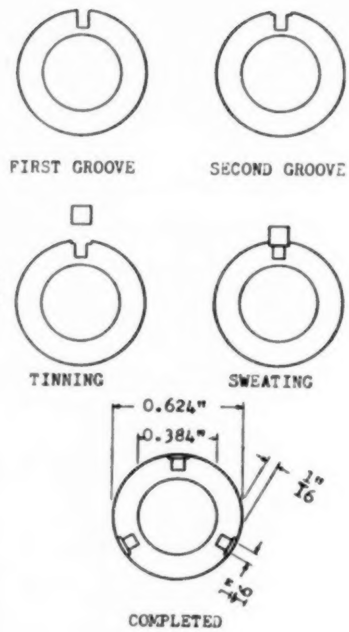

Fig. 2 Fabrication of Heat-Transfer Surface

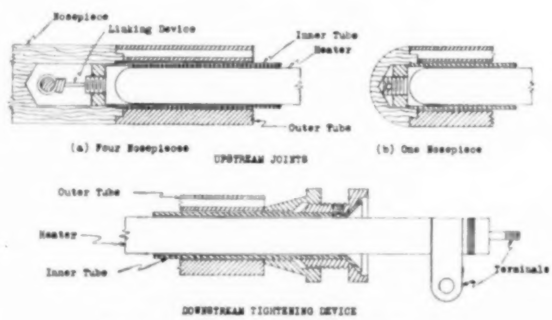

Fig. 3 Heat-Transfer Element Details

An impact tube mounted upon a sleeve which was slipped over the heating element served as a centering device. The element was judged to be centrally located in the air stream when rotation of the impact tube caused no change in its indication.

Vibrations of the nosepieces were measured using a motion-picture projector whose tachometer gave the frequency, while the amplitudes were obtained from shadows projected on a screen.

Test Procedures

In order to determine the uniformity of heat generation, the Calrod heater was placed horizontally in still air and surface temperatures were measured at constant power input. A chromel-alumel thermocouple, number 40 B and S gage, having a butt-welded hot junction, was suspended astraddle the heater, the hot junction being in good contact with the surface. Measurements were taken at intervals of 1 in. along the heater. An exact surface temperature may not have been obtained but any errors in the absolute values were likely to apply to all measurements.

The local excess above ambient temperature was assumed to be proportional to the local heat generation. It was possible to represent these reasonably well as a linear function of the length and to correct accordingly for the small deviations from uniform heat generation. The maximum observed deviation from the linear function was 2.9 per cent, the probable error being 1.1 per cent.

Calibration of the thermocouples in a hypsometer showed differences among them of 0.1 F with a maximum deviation from the computed steam temperature of 0.3 F.

The impact tube in the duct was calibrated, with the heating element in place, by a pitot tube of Prandtl's design with which traverses in the jet were taken at different velocities. A central core, 4 in. diam, suffered a velocity loss of 2 per cent at 43 in. downstream from the nozzle.

The experiments were classified into seven series of runs, given in Table 1.

TABLE 1 PLAN OF EXPERIMENT

Series designation	Nosepiece length, in.	Angle to stream, deg	Vibration controls
A	0.49	0	Yes
B	5.43	0	Yes
C	12.24	0	Yes
D	18.24	0	Yes
E	24.18	0	Yes
X	5.43	3	Yes
V	5.43	0	No

Notes:
1 Distance along a meridian from tip of nose to its joint with outer tube is called nosepiece length.
2 Angle between jet axis and longitudinal axis of heating element is called angle to stream.
3 No vibration controls means that dynamic vibration absorber and rubber shock mounts were removed.

The power input for a run was chosen as large as possible within the limits in which conduction losses by the thermocouples were negligible. With this in mind, the current was adjusted so that the temperature gradient along the heating element at the nosepiece did not exceed $1^1/_2$ F per in., being 1 F per in. or less for most of the runs.

Each run started with an exploration of the temperature distribution along the heating element at intervals of 2 in. Typical results of such explorations, for nearly extreme conditions, are shown in Fig. 4.

The probable temperature drop across the outer tube wall at the maximum heat input rate was less than 0.1 F. Since the measured temperature was between that of the inner and outer surfaces of this tube, no correction was applied and the measured temperature was taken as that of the outer surface.

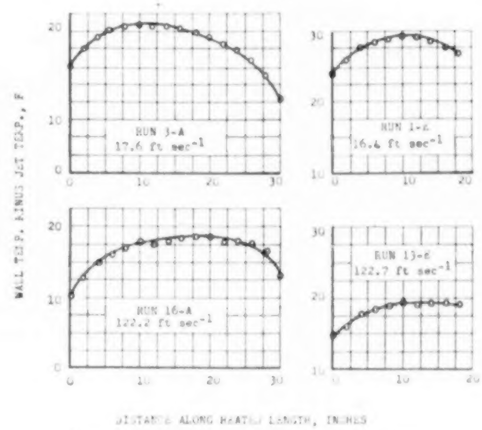

Fig. 4 Temperature Distribution Data

The present construction of the outer tube is geometrically similar to that of Jakob and Dow. They have shown that distortions of the temperature field near the holes in the wall can be neglected in this case.

The heat flow by convection q may be computed from the heat balance as follows

$$q = q_e - q_n - q_d - q_r \qquad [1]$$

where q_e is the electrical heat energy released in the thermal length x counted from the joint between nosepiece and outer tube, q_n is the heat loss to the nosepiece by conduction, q_d is the heat passing by conduction downstream from x, q_r is the heat loss by radiation to the surroundings. Calculation of q_n and q_d is based upon the assumption that the temperature gradients along the outer tube, inner tube, and heater unit are the same, although these are at different temperatures. Calculations under this assumption show that approximately 84 per cent of the heat conducted along the whole element may be conducted by the outer tube. The assumption involves least error as the gradient along the outer tube approaches zero. In the present experiment the loss q_n was almost the same for all runs, while q_d could be positive, negative, or zero.

Calculation of q_r showed that the equivalent surface coefficient by radiation was always nearly 0.1 B hr^{-1} ft^{-2} F^{-1}.

A mean surface coefficient of heat transfer h_m may be defined by

$$q = h_m S \theta_m \qquad [2]$$

where S is the surface area and θ_m is a mean temperature difference. Substituting from Equation [1] and introducing the circumference C and heating length x yield

$$h_m = \frac{q_e - q_n - q_d - q_r}{C x \theta_m} \qquad [3]$$

The term θ_m is defined as

$$\theta_m = \frac{1}{x}\int_0^x \theta \, dx \qquad [4]$$

This integration was performed with the aid of Simpson's rule with evaluation at intervals of 4 in. along the heated length.

Though the influence of aerodynamic heating may have reached 1 F, it was neglected because the relative velocity between the heating element and the thermocouples for air temperature was zero and both have cylindrical shapes.

Experimental Results

The range of experimental data is given in Table 2.

TABLE 2 RANGE OF EXPERIMENTS

Item	Symbol	Unit	Minimum	Maximum
Starting length	s	in.	0.49	24.18
Thermal length	x	in.	4	28
Total length	L	in.	4.49	40
Length ratio	x/L	1	0.141	0.983
Air velocity	v_0	ft/sec	9.9	123.6
Air temperature	t_0	deg F	62.7	84.3
Total heat input	q_t	B/hr	23.6	199.5
Coefficient of heat transfer	h_m	B/hr ft²F	2.81	26.6
Reynolds number	$(N_{Re})_L$	1	26100	2450000
Nusselt number	$(N_{Nu})_x$	1	60	3030

The total length L is defined as the distance along a meridian from the tip of the nose to a point downstream. The hydrodynamic starting length becomes $s = L - x$.

Considering that formation of boundary layers extends over the total length, the Reynolds number is defined as

$$(N_{Re})_L = \frac{v_0 L}{\nu_0} \qquad [5]$$

where v_0 and ν_0 are velocity and kinematic viscosity of the jet air, respectively.

The Nusselt number may be defined as

$$(N_{Nu})_x = \frac{h_m x}{k} \qquad [6]$$

where h_m is the mean coefficient of heat transfer over the thermal length x, and k is the thermal conductivity of the air at the film temperature $t_f = t_0 + \theta_m/2$.

The vibration data are given in Table 3.

TABLE 3 VIBRATION DATA

Number of runs	Air velocity, fps minimum	Air velocity, fps maximum	Amplitude at nosetip, in.	Frequency, rpm
		Series V Runs		
2	13.2	19.1	1/16	440
3	26.2	48.6	0	...
3	67.7	120.8	1/8	440
		Series B Runs		
5	9.9	22.8	1/16	520
4	28.2	44.3	1/32	520
7	55.5	122.2	1/16	520

The experimental results for the runs of Series A, B, C, D, and E are shown in Figs. 5 to 11, inclusive. Each gives the results for a fixed thermal length and variable starting length. For the higher Reynolds numbers the points could be connected by straight lines. Accordingly, turbulent boundary layers may have been obtained for Reynolds numbers from 300,000 to 600,000.

The influences of vibration and oblique crossflow are shown in Figs. 12 and 13, respectively, for a thermal length of 16 in. A comparison of these influences is shown in Fig. 14. Virtually the same influences were found for all thermal lengths.

According to Fig. 14, no net influence of vibration on heat transfer could be discovered in our experiments. It seems that the laminar sublayer was not affected by the vibrations.

The effect of oblique crossflow, on the other hand, was marked. The data could be represented reasonably well by two straight lines with different slopes as shown in Fig. 13. The Reynolds number of the point of intersection of these lines shifted from 150,000 to 600,000 as the thermal length was varied from 4 in. to 24 in. In all cases the slope of the steeper line corresponded to an exponent of 1.0, and the slope of the flatter line to an exponent of 0.57.

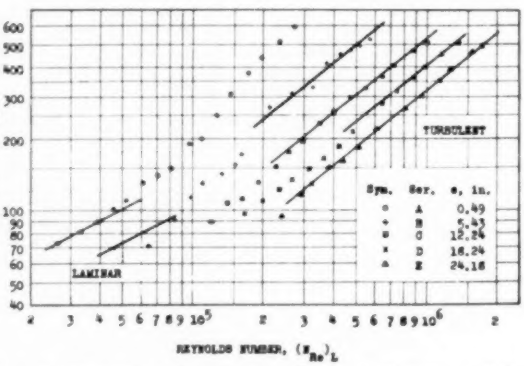

Fig. 5 Experimental Results for Thermal Length of 4 In.

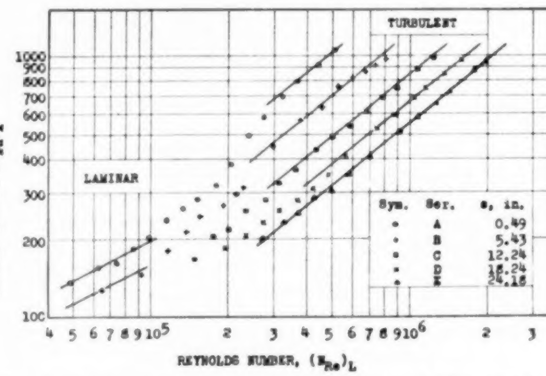

Fig. 6 Experimental Results for Thermal Length of 8 In.

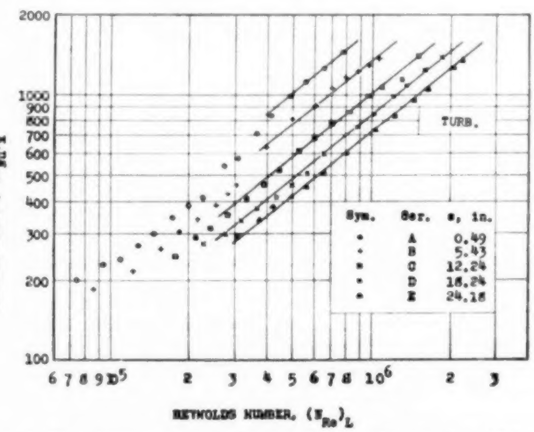

Fig. 7 Experimental Results for Thermal Length of 12 In.

Correlation of Data

Each straight line for the turbulent ranges shown in Figs. 5 to 11, inclusive, could be represented[5] in the form

[5] Exponents larger than 0.8, sometimes reported in the literature, may have been obtained because of imperfect parallel flow. As shown in the previous section, a deviation by 3 angular deg was sufficient to increase the exponent markedly.

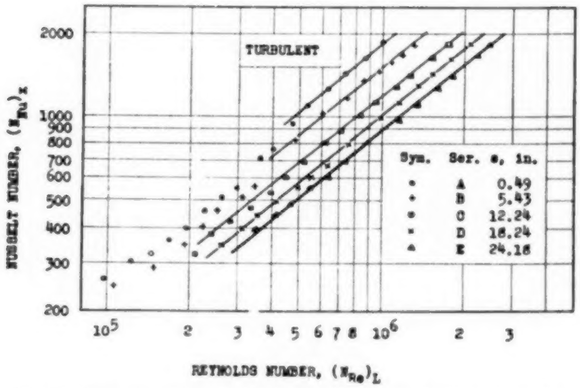

FIG. 8 EXPERIMENTAL RESULTS FOR THERMAL LENGTH OF 16 IN.

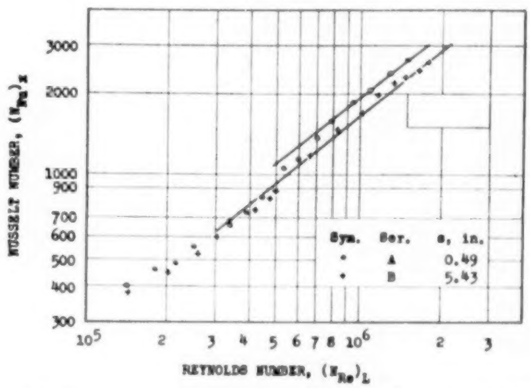

FIG. 10 EXPERIMENTAL RESULTS FOR THERMAL LENGTH OF 24 IN.

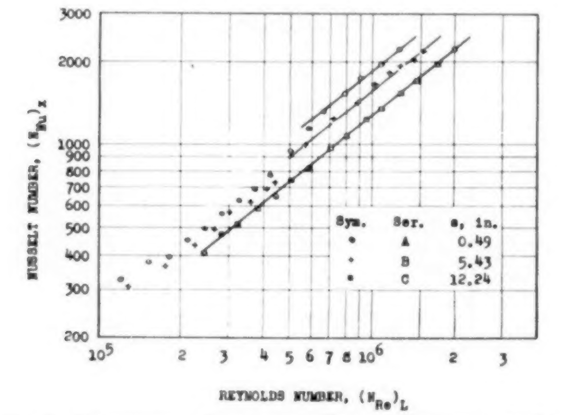

FIG. 9 EXPERIMENTAL RESULTS FOR THERMAL LENGTH OF 20 IN.

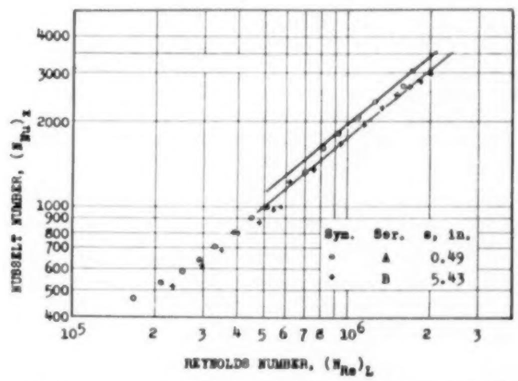

FIG. 11 EXPERIMENTAL RESULTS FOR THERMAL LENGTH OF 28 IN.

$$(N_{Nu})_x = F_x(N_{Re})_L{}^{0.8} \quad \ldots\ldots [7]$$

where F_x is a function of x/L.

The function is plotted in Fig. 15. Each point represents one of the lines in Figs. 5 to 11. The points are lying on a straight line which could be represented by

$$F_x = 0.0307\left(\frac{x}{L}\right)^{0.91} \quad \ldots\ldots [8]$$

Combination of Equations [7] and [8] yields

$$(N_{Nu})_x = 0.0307\,(N_{Re})_L{}^{0.8}\left(\frac{x}{L}\right)^{0.91} \quad \ldots\ldots [9]$$

Examination of Fig. 15 shows this to be a well-founded correlation. For x/L between 0.25 and 0.40, there is an overlap between the Series C, D, and E runs, respectively. For x/L of 0.6, there is an overlap between the Series B and C runs, respectively. This seems to indicate that the thermal boundary layer which starts at $x = 0$ reaches the thickness of the hydrodynamic layer almost immediately.

The final correlation is given in Fig. 16. It is seen that effective correlation was obtained in the turbulent range for Reynolds numbers greater than 500,000. The results of Jakob and Dow are shown for comparison purposes. The points lying below Jakob and Dow's line for the laminar range are based on very low velocities which could not be measured exactly.

Table 4 is presented for comparison of correlations in the turbulent range with the present results.

TABLE 4 COMPARISON OF CONFIGURATION FUNCTIONS IN CORRELATIONS OF MEAN SURFACE COEFFICIENTS AT TURBULENT BOUNDARY LAYERS

x/L	Present work	Jakob and Dow	Maisel and Sherwood	s/L
0.0	(∞)	(1.400)	(∞)	1.0
0.1	1.230	(1.299)	1.318	0.9
0.2	1.157	(1.216)	1.220	0.8
0.3	1.114	(1.150)	1.166	0.7
0.4	1.086	1.098	1.128	0.6
0.5	1.064	1.059	1.098	0.5
0.6	1.048	1.032	(1.075)	0.4
0.7	1.032	1.015	(1.057)	0.3
0.8	1.020	1.005	(1.036)	0.2
0.9	1.010	1.001	(1.019)	0.1
1.0	1.000	(1.000)	(1.000)	0.0

NOTE: Interpolated values in parentheses.

Multiplying each side of Equation [9] by L/x yields

$$\left.\begin{aligned}(N_{Nu})_L &= \frac{L}{x}\,(N_{Nu})_x \\ &= 0.0307\,(N_{Re})_L\left(\frac{x}{L}\right)^{-0.09} \\ &= 0.0307\,(N_{Re})_L\,\phi_1\end{aligned}\right\} \quad \ldots\ldots [10]$$

where the configuration function

$$\phi_1 = \left(\frac{x}{L}\right)^{-0.09} \quad \ldots\ldots [11]$$

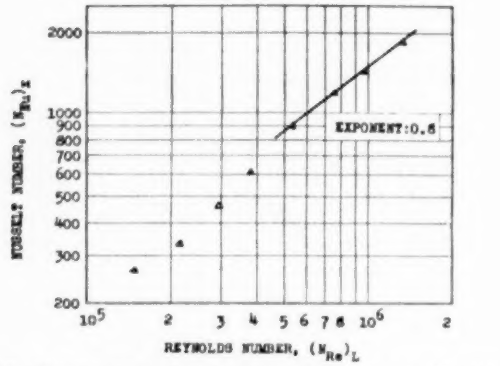

FIG. 12 EXPERIMENTAL RESULTS WITH VIBRATION FOR THERMAL LENGTH OF 16 IN. (SERIES V RUNS)

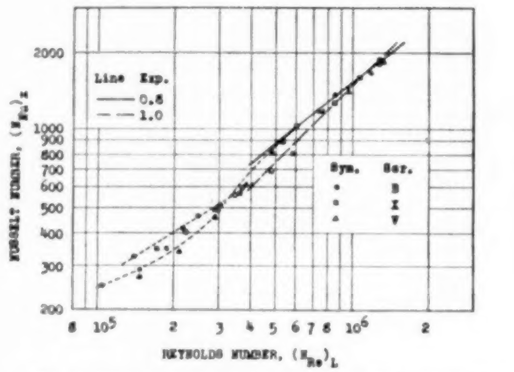

FIG. 14 COMPARISON OF VIBRATION AND OBLIQUE CROSSFLOW DATA FOR THERMAL LENGTH OF 16 IN.

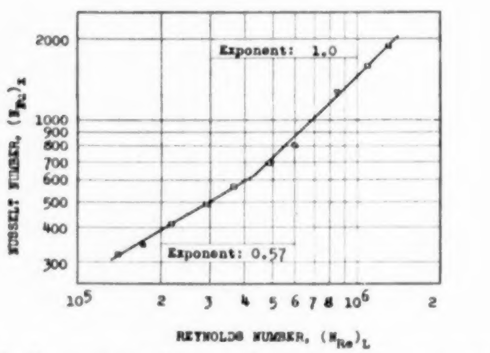

FIG. 13 EXPERIMENTAL RESULTS WITH OBLIQUE CROSSFLOW FOR THERMAL LENGTH OF 16 IN. (SERIES X RUNS. ELEMENT AXIS AT 3 DEG TO JET AXIS)

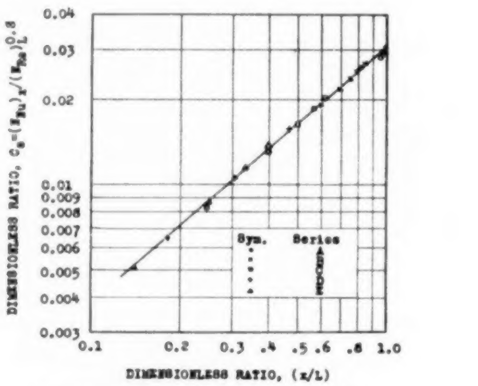

FIG. 15 SURFACE CONFIGURATION DATA

This corresponds to the function

$$\phi_2 = 1 + 0.40\left(\frac{s}{L}\right)^{2.76} \qquad [12]$$

as used by Jakob and Dow, and

$$\phi_3 = \left[1 - \left(\frac{s}{L}\right)^{0.8}\right]^{-0.11} \qquad [13]$$

as used by Maisel and Sherwood.

Jakob and Dow's data were obtained for values of x/L between 0.4 and 0.9, while the present work covers values between 0.14 and 0.98. Extrapolated data in Table 4 are identified by parentheses.[4] Examination of the data in Table 4 shows excellent agreement between the previous and present work in the range studied by Jakob and Dow.

LOCAL COEFFICIENTS

The heat flow between a surface and a fluid may be written as

$$q = \int_0^x h\theta\, dx = h_m\theta_m x \qquad [14]$$

where h and θ are local values.

Differentiation yields

$$h = \frac{1}{\theta}\frac{d}{dx}(h_m\theta_m x)$$

$$= h_m\frac{\theta_m}{\theta} + h_m\frac{x}{\theta}\frac{d\theta_m}{dx} + x\frac{\theta_m}{\theta}\frac{dh_m}{dx} \qquad [15]$$

Differentiation of θ_m, as defined by Equation [4], yields

$$\frac{d\theta_m}{dx} = \frac{\theta - \theta_m}{x} \qquad [16]$$

Equation [9] may be generalized and rewritten as

$$\frac{h_m}{C_0\left(\frac{v_0}{\nu_0}\right)} = k(s+x)^{m-n}(x)^{n-1} \qquad [17]$$

which is considered as a function of x alone.

Differentiation of Equation [17] and substitution from Equations [17] and [16] into [15] yield

$$\frac{h}{h_m} = 1 + \frac{\theta_m}{\theta}\left[\frac{\theta - \theta_m}{k}\frac{dk}{d\theta_m} + n - 1 + (m-n)\frac{x}{s+x}\right] \qquad [18]$$

The conditions during the experiments were such that the term $[(\theta - \theta_m)/k]dk/d\theta_m$ was negligible. Substituting the previously determined values $m = 0.8$ and $n = 0.91$ into Equation [18] yields

[4] Jakob and Dow's definition of the starting length differs somewhat from the present one. Differences in the results were found to be negligible.

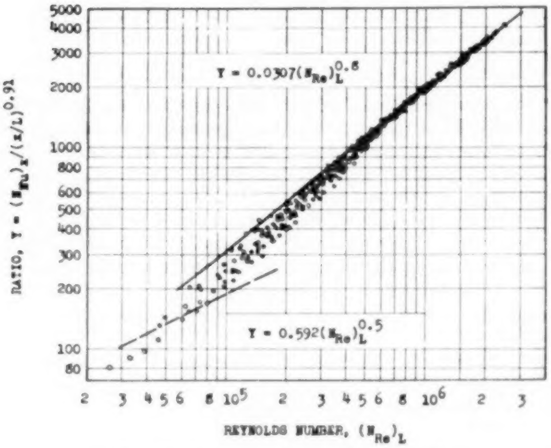

FIG. 16 CORRELATION OF ALL RESULTS
(Jakob and Dow's results for laminar flow are shown for comparison.)

$$\frac{h}{h_m} = 1 - \frac{\theta_m}{\theta}\left(0.09 + 0.11\,\frac{x}{L}\right) \qquad [19]$$

Equation [19] was tested with experimental data in which h was calculated for 2-in. lengths of the heater element, the end losses being calculated from temperature-distribution curves similar to Fig. 4.

The experimental results were greater than the calculated results from 0 to 12.9 per cent of the experimental values, the most serious discrepancy, 12.9 per cent, occurring for x/L = 0.98. For x/L smaller than 0.95 the agreement between experimental and calculated results was within 5 per cent of the experimental value for 80 per cent of the calculations.

Considering that differentiation of an empirical relation was used for the comparison of the local coefficients from the basic data, the agreement between experimental and calculated results was very satisfactory.

For an isothermal surface the following Nusselt number for local coefficients was derived

$$(N_{Nu})_{x,\,loc} = (N_{Nu})_x\left(0.91 - 0.11\,\frac{x}{L}\right)$$

or

$$(N_{Nu})_{x,\,loc} = 0.0246(N_{Re})_L^{0.8}\left[1.138\left(\frac{x}{L}\right)^{0.91} - 0.138\left(\frac{x}{L}\right)^{1.91}\right] \qquad [20]$$

and

$$(N_{Nu})_{L,\,loc} = 0.0246(N_{Re})_L^{0.8}\left[1.138\left(\frac{x}{L}\right)^{-0.09} - 0.138\left(\frac{x}{L}\right)^{0.91}\right] \qquad [21]$$

The constants in these equations have been rearranged so that the configuration function in brackets has the value of unity for $x/L = 1$.

Rearrangement of constants of other investigations permits the comparison shown in Table 5. The present results and those of Jakob and Dow are obtained by differentiation, Rubesin's from theoretical deduction, and Maisel and Sherwood's (7) from diffusion experiments. The configuration function for local

TABLE 5 COMPARISON OF CONFIGURATION FUNCTIONS ACCORDING TO EQUATION [20] WITH THOSE OF PREVIOUS INVESTIGATORS

x/L	Rubesin	Maisel and Sherwood	Jakob and Dow	Present work	x/L
0.0	∞	(∞)	(1.750)	(∞)	1.0
0.1	1.518	1.451	(1.490)	1.383	0.9
0.2	1.341	1.323	(1.311)	1.281	0.8
0.3	1.246	1.253	(1.200)	1.223	0.7
0.4	1.183	1.198	1.128	1.176	0.6
0.5	1.136	1.154	1.090	1.138	0.5
0.6	1.099	(1.117)	1.069	1.103	0.4
0.7	1.069	(1.083)	1.053	1.077	0.3
0.8	1.042	(1.053)	1.042	1.049	0.2
0.9	1.021	(1.026)	1.025	1.023	0.1
1.0	1.000	(1.000)	(1.000)	1.000	0.0

NOTE: Interpolated values in parentheses.

mass transfer which was used to obtain the values of Maisel and Sherwood in Table 5, was derived by differentiation of their empirical equation for mean mass transfer. Extrapolated values are in parentheses. The agreement of the present results with those of other investigators is very satisfactory.

While the surface temperature was not perfectly constant over the portion of the heating length considered in the computations, the gradient was not large at the worst and almost equal to zero for considerable portions of the length. Hence the results of our experimentation should not deviate much from those for uniform surface temperature. However, some irregularities and scatterings seen in Figs. 5 to 11, and 16 may be due to the temperature variation along the heated length.[7]

INFLUENCE OF SURFACE CURVATURE

Previous investigations, reduced to zero starting length, can be represented by the form

$$(N_{Nu})_x = B_0(N_{Re})_x{}^n \qquad [22]$$

because at zero starting length $(N_{Nu})_x = (N_{Nu})_L$ and $(N_{Re})_x = (N_{Re})_L$.

The results of previous investigations, taken from Jakob and Dow's paper, and the present results are given in Table 6.

TABLE 6 VALUES OF B_0 AND EXPONENT n OF EQUATION [22] FOR TURBULENT BOUNDARY LAYERS

Author	Kind of work	Surface	B_0	n
Latzko	Theoretical[a]	Plane	0.0253	0.8
Juerges	Experimental	Plane	0.0322	0.8
Latzko	Theoretical[b]	Plane	0.0317	0.8
Colburn	Correlation	Plane	0.0320	0.8
Seibert	Theoretical	Plane	0.0349	0.8
Jakob and Dow	Experimental	1.3 in. diam cyl	0.0280	0.8
Tessin and Jakob	Experimental	0.624 in. diam cyl	0.0307	0.8

[a] As interpreted by Jakob and Dow.
[b] As interpreted by Tessin and Jakob.

Jakob and Dow showed that in order to obtain values of heat transfer between a fluid and a cylinder, one may multiply those for a flat plate by a factor, which for turbulent boundary layers is

$$F = 1 + 0.3\,\frac{b}{r} \qquad [23]$$

where b is the boundary-layer thickness, and r is the radius of the cylindrical cross section. Thus for a flat plate $F = 1$.

Using a linear function of $1/r$ as in Equation [23], and combining the values of B_0 and r according to Jakob and Dow's and the present work, the following is obtained

[7] A reviewer has invited our attention to more recent results of Rubesin (8), who shows that the local heat-transfer coefficient on a plate having constant heat-transfer rate differs from that on a plate at a constant temperature by only 6 per cent.

$$B_0 = 0.0255 + 1.35(10)^{-4}\frac{1}{r} \qquad [24]$$

where r is taken in feet units.

Extrapolation of Equation [24] to $r = \infty$ (the case of the flat plate) yields $B_0 = 0.0255$. From Table 6 it is seen that Latzko's theoretical value for the flat plate as interpreted by Jakob and Dow is 0.0253. They converted Latzko's equation into the form

$$N_{Nu} = 0.0356(N_{Re})^{0.80}N_{Pr} \qquad [25]$$

noticing that this equation was derived under the assumption that $N_{Pr} = 1$. Then they assumed that the equation may approximately hold for $N_{Pr} = 0.71$ (for air). The result is shown by the dotted line in Fig. 17.

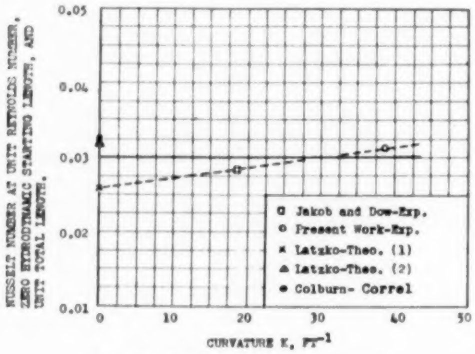

Fig. 17 Influence of Curvature

(1, As interpreted by Jakob and Dow; 2, As interpreted by Tessin and Jakob.)

Another interpretation would be that for $N_{Pr} = 1$, Latzko's equation converts to

$$N_{Nu} = 0.0356(N_{Re})^{0.80} \qquad [26]$$

Now there is evidence from empirical data as shown, for instance in Fig. 20 of Colburn's (5) paper, that

$$N_{Nu} = C_0(N_{Re})^{0.8}(N_{Pr})^{m} \qquad [27]$$

Colburn correlated data for flat plates with $m = 1/3$. Accepting this, comparison of Equations [26] and [27] yields

$$N_{Nu} = 0.0317(N_{Re})^{0.8}$$

for air streaming parallel to a flat plate. According to this interpretation of Latzko's equation, $B_0 = 0.0317$ for $r = \infty$.

For the time being, the constant $B_0 = 0.03$ independent of curvature, in the range investigated, seems to be the most probable result. This is shown by the full line in Fig. 17.

Therefore it is recommended that the following equation for air in the range $r = 0.3$ in. to $r = \infty$ be used

$$(N_{Nu})_x = 0.03(N_{Re})_L^{0.8}\left(\frac{x}{L}\right)^{0.91}$$

and for other fluids

$$(N_{Nu})_x = 0.034(N_{Re})_L^{0.8}(N_{Pr})^{1/3}\left(\frac{x}{L}\right)^{0.91}$$

The differences of the values of other observers cannot be explained at this time. It may be that different levels of turbulence in the arrangements of the various observers have caused this discrepancy. Continuation of the experiments with this in mind is planned.

Summary and Conclusions

1 A cylindric heat-transfer element, 0.624 in. diam, and various wooden nosepieces with hemispherical tips to be exposed to air streaming parallel to the axis were constructed. The ratio of thermal length to total length could be varied from 0.141 to 0.983.

2 In the experiments Reynolds numbers from $2.6(10)^4$ to $2.4(10)^6$ were obtained with transition starting at Reynolds number of 50,000 and ending from 300,000 to 600,000.

3 A correlation for Nusselt number based on mean surface coefficients was obtained, which included the effects of starting length.

4 An equation for Nusselt number based on local surface coefficients was derived, which was in satisfactory agreement with the experimental results.

5 Vibration and oblique crossflow studies were made to show that the present results were not influenced by these factors.

6 Comparing the results of a previous investigation by Jakob and Dow with the present results indicates a decrease of the heat transfer by 8.8 per cent if the cylinder diameter is increased from 0.624 in. to 1.3 in. Extrapolation to infinite diameter (flat plate), however, is uncertain and, therefore, it is recommended that an average value for the heat-transfer coefficient be used, independent of the curvature until further evidence of the influence of curvature and turbulence level is available.

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Discussion

D. S. Maisel.[8] This paper represents an extension of previous work carried out at the Illinois Institute of Technology on the transfer characteristics from a surface, where conditions for momentum transfer differ from those for mass or heat transfer. These have been concerned with the influence of a "starting section" over which momentum transfer alone occurs on the

[8] Esso Laboratories, Development Division, Standard Oil Development Company, Linden, N. J.

heat and mass transfer rates from the downstream transfer surface.

The authors conclude that the well-founded correlation between ratio of the length of the heat-transfer surface to the sum of the lengths of the starting section plus the transfer section means that the "correlation is independent of the absolute lengths of the thermal or starting portions and it seems to indicate that the thermal boundary layer reaches the thickness of the hydrodynamic layer almost immediately." This is very interesting since it would seem appropriate to believe that the two lengths should be independently important. The point of conversion from a laminar to a turbulent zone either in the starting length or heat-transfer surface, for example, should seem to affect the heat-transfer rates.

The various configuration functions shown by the authors in Table 2 do not seem to differ by any great degree. Considering errors inherent in measurement, each, with the possible exception of the Jakob and Dow function for small values of (X/L) should correlate data equally well. Therefore there may be some preference to use the form suggested by Rubesin, which is based on a theoretical analysis of the transfer dynamics.

Authors' Closure

We agree with Dr. Maisel's statement that the rates of heat transfer should be different according to whether heating starts at a Reynolds number below or above the critical point. We might have mentioned that all points from which Equation [9] was obtained, were taken from experiments in which s was larger than the critical distance s_{cr} at which turbulence started. Our equations of correlation do not cover the transient region where heating may have started below or above the critical Reynolds number; in this region the points for Y do not fall in a unique line, as can be seen from Fig. 16 at $(N_{Re})_L \leq 500{,}000$. In addition to the ratio x/L a ratio $(s_{cr} + x)/L$ or any other length ratio, including s_{cr}, might enter the correlation. It would be difficult to determine s_{cr} with reasonable accuracy. Anyway, we did not find any influence of s_{cr}.

The first part of the passage, verbally quoted in Maisel's discussion, was included in the preprint of our paper, but was omitted as self-explanatory in the final draft; obviously, the right side of Equation [9] is independent of the absolute value of x or L-x, but dependent on their relative values x/L or $(L$-$x)/L$.

The second part of that quotation refers to the high exponent (0.91) of x/L. If the exponent were 1.0, then h_m would be independent of x. This means that the thermal boundary layer over the heating length would have the same mean thermal resistance, at whatever place of the hydrodynamic boundary layer heating were started. If one could assume that the resistance of the buffer layer were negligible compared with that of the laminar sublayer and the thickness of the sublayer did not change with x, then the mean thermal resistance would be the same for any value $s \gtreqless s_{cr}$; however, since the exponent is not 1, but only 0.91, the resistance need not be exactly the same. This assumption may be closer to reality than that made in our paper and quoted by Dr. Maisel.

The Determination of Local Forced-Convection Coefficients for Spheres

By JOHN R. CARY,[1] LONG BEACH, CALIF.

The local heat-transfer coefficient on the surface of a sphere is investigated in the range of Reynolds numbers from 44,000 to 151,000. Correlation of experimental data is based on the Nusselt and Reynolds numbers, and two curves are presented for the relation between the Nusselt and Reynolds numbers as a function of the angle from the stagnation point.

Introduction

In the course of the design of an airplane component consisting of a large hemispherical shell required to be maintained free of ice, a survey of the literature revealed no data for predicting the local rate of heat dissipation from a spherical surface exposed to the cooling influence of a surrounding medium moving relative to the sphere.

Since there existed an immediate need for these data, an experimental program to determine the value of the local forced-convection coefficient was initiated.

Although the study was carried out on a sphere, it is anticipated that the data obtained may be applied as first approximations to local coefficients on geometrically similar bodies such as paraboloids, ellipsoids, and other surfaces of revolution until such time as data for these particular surfaces may be gathered.

Description of Apparatus

The test apparatus shown in Fig. 1 consisted of a 5-in-diam Armco-iron hollow sphere containing a thermally isolated heating element, and mounted concentrically in a 15-in-diam aluminum duct; an air-supply system consisting of a centrifugal blower driven by an electric motor operating on 110 volts alternating current; a butterfly valve to control air velocity in the duct; a small boiler for generating steam; a wet cell for supplying electric power to the local heating element; a microrheostat for controlling power in the heating element; and instruments for measuring pressure and temperature.

The duct was assembled from sections connected at flange joints, and was approximately 40 ft long. The butterfly valve was located immediately downstream of the blower discharge. The length of duct between the valve and the sphere was 18 ft, 30 in. of which were straightening vanes located a short distance upstream of the sphere. Distances of 9 in. and 13 in., respectively, upstream from the sphere were a pitot tube for determining air velocities and a mercury-in-glass thermometer for determining duct-air temperatures.

Air at ambient temperature was used as a coolant. Owing to the quantity of air required (up to nearly 7 lb per sec at maximum velocity), no attempt was made to vary air temperature by refrigerating the air supply. Heating of the air was considered and discarded in order that large temperature differences would exist between the duct air and the heater element.

[1] Design Engineer, Douglas Aircraft Company.

Contributed by the Heat Transfer Division and presented at the Fall Meeting, Chicago, Ill., September 8–11, 1952, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, June 20, 1952. Paper No. 52—F-29.

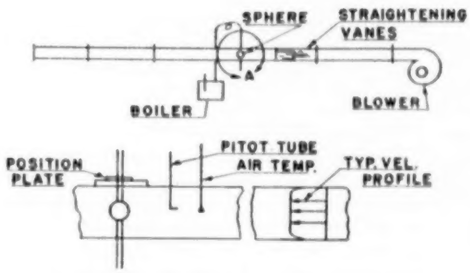

Fig. 1 Schematic Diagram of Apparatus

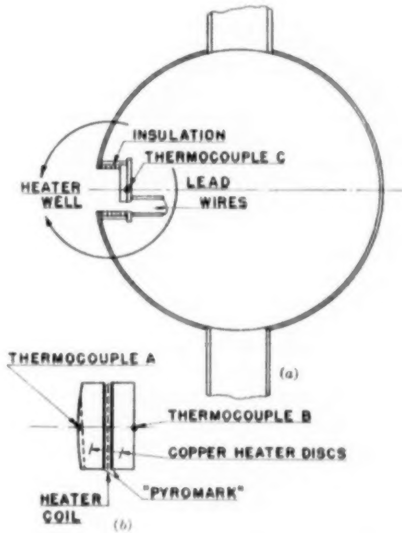

Fig. 2 Sketch of Sphere and Heater Sandwich

The spherical shell shown in Fig. 2(*a*) was a hollow Armco sphere of 5 in. diam, with a wall thickness of 0.040 in. A recess was constructed in the sphere to contain a local heating element, which was inserted in the recess flush with the surface of the sphere, in order that the spherical contour would remain unchanged. The heater unit, Fig. 2(*b*), was assembled in such a way that the heater would be nearly isolated thermally from the rest of the sphere, so that all heat generated in the heating element could escape only to the cooling-air stream.

The heater proper was formed by making a sandwich of two copper disks between which was located a heater coil of nichrome wire. The surfaces of the heater disks adjacent to the coil were painted with "Pyromark" electrical insulating paint. A chromel-alumel thermocouple was installed on each face of the assembled sandwich, as well as one on the floor of the heater well. The external surface thermocouple, shown as thermocouple *A* in Fig.

2(*b*), indicated the temperature of the surface exposed to the coolant, while the interior thermocouples *B* and *C* were intended to demonstrate the achievement of a thermally insulated condition for the rear surface of the heater.

The thermocouples were placed in shallow grooves cut into the surface, and the grooves were filled with solder to assure a smooth spherical contour. It has been shown (1)[2] that marked effects on the heat-transfer coefficient are observed if the boundary layer is disturbed by any small irregularity of surface.

Steam generated locally in a small boiler was fed to the inside of the sphere to maintain a nearly uniform temperature on the surface of the sphere. A second consideration was that of obtaining a high degree of insulating effect on the inner surface of the heater sandwich. Reference to Fig. 2(*b*) shows that if sufficient electric power is supplied to the heater to maintain the inner surface of the electrically heated sandwich containing thermocouple *B* at the same temperature as the adjacent steam-heated surface of the heater-well floor, containing thermocouple *C*, then none of the electrically generated heat may flow inward. On the other hand, heat may not flow from the steam to the heater sandwich. Therefore all controlled heating due to electric power must flow to the cooling-air stream, permitting a heat balance to be obtained.

It also will be noted that, should the temperature of the sphere be somewhat different from the temperature of the heater sandwich, the thin wall of the sphere permits very little heat to flow by direct conduction, and the effect is further minimized by the insulation shown in Fig. 2(*a*). In operation, temperatures *B* and *C* were not measured absolutely, but merely maintained equal by adjustment of the power supply to the heating element.

Electric power for the heating element was drawn from a 2-volt wet cell, and the voltage in the heater controlled by the microrheostat.

The sphere was mounted in the duct so that it was free to rotate 360 deg about the vertical axis. A positioning plate was attached to the duct so that a scribed line on the rotating axis could be used to mark the position of the heating element relative to the stagnation point of the sphere in the air stream. Means were provided for clamping the sphere in position to preclude the possibility of inadvertent rotation. The vertical axis was a length of 1-in-diam seamless steel tubing, which also served as a conduit for the thermocouple and power leads to the sphere.

Determination of Preliminary Data

As a means for determining air velocity in the duct, the use of a Pitot-tube probe was anticipated because of the physical size of the duct. In order that pitot-tube readings would reflect accurately the air velocity, a series of runs were made at each setting of the butterfly valve to permit traversing the duct with the Pitot tube. The velocity profile across the duct was found to be very nearly straight across 85 per cent of the duct diameter. A typical velocity profile is shown in Fig. 1. Since the diameter of the sphere was only one third the diameter of the duct, the velocity profile of the air approaching the sphere was linear across the projected diameter of the sphere. Accordingly, the air velocity used in the correlation of experimental data was taken as the rectilinear maximum.

Owing to the presence of the sphere in the duct, a constriction of the flow area occurred. However, even at the diameter of the sphere, the flow area was 89 per cent of the area in the free duct. Somewhat higher velocities were to be expected at the diameter of the sphere, but these were not expected to be especially important because of the relatively small restriction. Stream lines computed from the potential function showed rather little distortion at a distance of slightly more than one diameter from the center of the sphere.

The heater sandwich was calibrated by direct measurement of the electrical resistance. The resistance of the assembled circuit, including the heater coil, 122 in. of 18-gage copper wire in the leads, and the contact resistance at a voltmeter was measured with an impedance bridge and found to be 0.906 ohm. Reference to published data gives the resistance of the copper leads as 6.51 ohms/1000 ft at 77 F. The resistance of the leads was, therefore, 0.067 ohm. The contact resistance was determined to be 0.117 ohm. The heater resistance was, therefore, 0.722 ohm.

The heat generated in the heater coil could then be determined as follows: For a circuit containing the contact resistance, the lead-wire resistance, and heater resistance in series, the circuit current was

$$I = \frac{E}{R_c + R_l + R_h} = \frac{E}{\Sigma R}$$

Thus the heat generated in the sandwich was

$$W = I^2 R_h = \left(\frac{E}{\Sigma R}\right)^2 R_h = KE^2$$

where

$$K = \frac{R_h}{(R_c + R_l + R_h)^2} = 0.879 \text{ ohm}^{-1}$$

Procedure

With cooling air supplied by the blower at the desired velocity, steam was generated in the boiler and introduced to the interior of the sphere. Current drawn from the wet cell was used to generate heat in the isolated heater element. By means of the microrheostat, the voltage in the heater coil was controlled so that it would generate enough heat to keep the inward surface of the heater sandwich at the same temperature as the floor of the heater well, thus providing thermal equilibrium at this surface. The cooling-air temperature was read from a mercury-in-glass thermometer graduated in 1-deg intervals, and the temperature of the exposed surface of the heater sandwich obtained from thermocouple readings. Values of air temperature, heater-sandwich surface temperature, and angular location of the heater sandwich were recorded for each air velocity. The angular position of the heater sandwich was varied from 0–360 deg from stagnation in 15-deg increments. Air velocities from 22 fps to 73.3 fps were used, the lower limit being chosen because of the doubtful accuracy of velocity readings at much lower values, and the upper limit being imposed by the duct size and blower capacity.

The heat flux escaping from the heater sandwich was determined from

$$\phi = \frac{K\,E^2}{A} = \frac{(0.879)\,(3.41)E^2}{(0.441/144)} = 976\,E^2 \text{ Btu/hr-sq ft} \quad [1]$$

where the area A of the heater sandwich was 0.441 sq in. Also

$$\phi = h(t_s - t_0) + \sigma F\left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right] \quad [2]$$

The configuration factor F in the radiant component of the heat transferred was small owing to the low emissivity of the aluminum duct and the copper heater sandwich. The latter was maintained in a polished state by the use of emery cloth. The radiant-energy transfer varied between approximately 0.5 and 4.0 per cent of the total.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

The heat-transfer coefficient was determined from the equality of Equation [1] and Equation [2], namely

$$h = \frac{976\,E^2 - 0.00913\,[(T_s/100)^4 - (T_0/100)^4]}{(t_s - t_0)}$$

Btu/hr-sq-ft deg F . . . . [3]

The effect of aerodynamic heating was negligible in the speed range employed.

Theory

An analytical approach to the convection coefficient may be based on a boundary-layer study. Among the many contributions to the literature dealing with boundary-layer study, two seem to be most applicable to the present case. Millikan (2) was apparently the first to prove that the von Kármán momentum integral applies even near the stagnation point of a body of revolution. Tomotika (3) investigated the particular case of a sphere in a uniform stream. Using a quartic for the velocity profile in a laminar boundary layer, Tomotika evaluated the momentum integral, and expressed the boundary-layer thickness as a function of the angle from the stagnation point in terms of the free-stream Reynolds number and a parameter Z such that

$$\delta = \sqrt{\frac{Zd^2}{2N_{Re,0}}} \quad\text{. . . . . [4]}$$

Tomotika computed Z by two methods as follows:

(a) Using the local velocity about the sphere obtained from the potential theory, $V = V_0\,(3/2)\sin\theta$.

(b) Applying the results of an earlier pressure-distribution investigation by Flachsbart (4).

In both cases, Z increases with θ, with Z for the experimental data rising more steeply than Z for the potential theory velocity. In order to obtain a qualitative idea of the heat-transfer coefficient, it suffices to examine the function Z, since the heat-transfer coefficient varies inversely with the boundary-layer thickness δ, and therefore inversely with Z. Thus it was to be expected that the heat-transfer coefficient would decrease with increasing θ as long as the boundary layer remained laminar.

Pohlhausen (5) had shown that the heat-transfer coefficient for a fluid medium with Prandtl number of 1 could be expressed in terms of the laminar-boundary-layer thickness and a parameter dependent upon the shape of the velocity profile in the boundary layer. This expression is

$$\frac{h\delta}{k} = 0.765 \quad\text{. . . . . [5]}$$

Comparison with the broad treatment of boundary-layer literature outlined by Schlichting (6) gives assurance that the boundary-layer thicknesses according to Tomotika and Pohlhausen are the same. Therefore, applying Tomotika's expression, it is readily seen that the local heat-transfer coefficient may be expressed in terms of the Nusselt number, the Reynolds number, and the Z-function. Thus

$$\frac{N_{Nu}}{\sqrt{N_{Re,0}}} = \frac{0.765}{\sqrt{Z/2}} \quad\text{. . . . . [6]}$$

At the stagnation point, for example, where $Z = 3.144$

$$\frac{N_{Nu}}{\sqrt{N_{Re,0}}} = 0.61 \quad\text{. . . . . [7]}$$

Comparison of values computed by this method and experimental values obtained in the present investigation will be discussed later.

Johnstone and co-workers (7), by integrating the Boussinesq equation for heat transfer to a submerged flat plate over the surface of a sphere, arrived at an expression which may be reduced to the following

$$\frac{N_{Nu}}{\sqrt{N_{Re,0}}} = 0.614\,\sqrt{N_{Pr}} \quad\text{. . . . . [8]}$$

Details of the Johnstone development are not given, and it is not known whether particular attention was given to the possibility of separation and other related phenomena. Furthermore, it should be pointed out that Jakob (8) suggests that the Boussinesq equation is not valid because of oversimplified assumptions.

Discussion of Results

Numerical results of the present investigation are plotted in Fig. 3 which shows the coefficients decreasing for increasing θ in general agreement with theory. Each value in Fig. 3 represents an average of two values obtained at $+\theta$ and $-\theta$, respectively.

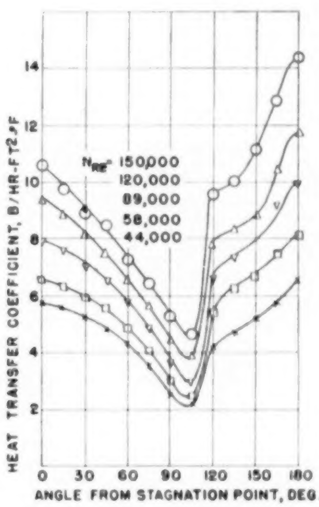

Fig. 3 Results of Investigation

A peculiarity of the curves merits some discussion. It will be noted that the coefficient decreases to a minimum at an angle of approximately 105 deg, and then rises precipitously, appearing to stabilize again at 120 deg. This is probably due to a transition from laminar flow to turbulent flow in the boundary layer, followed by separation of the boundary layer from the sphere, and a highly turbulent state in the region of the adverse pressure gradient on the rear surface of the sphere. It is probable that separation occurs very soon in an adverse pressure gradient, so that here separation possibly occurs soon after 100 deg.

It is of interest to note that Flachsbart (4) found that separation occurred at $\theta = 82$ deg, and Tomotika (3), using Flachsbart's pressure-distribution data, was able to compute the separation angle as 81 deg. In contrast to these values, the angle of separation computed by Tomotika for the potential theory velocity was 108 deg. It appears likely, then, that the separation angle may be influenced to an appreciable degree by the presence of ducting.

An effort was made to obtain further values of the heat-transfer coefficient in the interval 100 deg $\leq \theta \leq$ 120 deg. The values obtained were quite erratic, however, and are not included.

It was decided to correlate experimental data as indicated by the boundary-layer study. It was necessary, as might be anticipated, to establish two expressions, depending upon whether or not separation had occurred.

For values of $\theta \leq 105$ deg, the Nusselt number varies with the square root of the Reynolds number

$$N_{Nu} \alpha (N_{Re})^{0.50} \qquad [9]$$

in exact agreement with the boundary-layer theory, at least up to θ of approximately 80 deg. Fig. 4 shows the parameter $N_{Nu}/(N_{Re})^{0.50}$ plotted against the angle from the stagnation point for 0 deg $\leq \theta \leq$ 105 deg. Also shown are values computed using Tomotika's boundary-layer thickness, with Z computed from the data of Flachsbart. It will be seen that the experimental values exhibit less than 15 per cent departure from theoretical values, which is possibly to be considered a fortuitous circumstance.

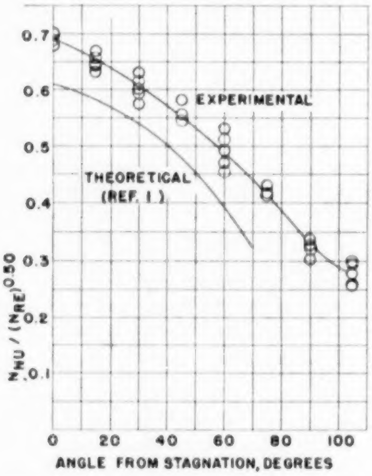

FIG. 4 CORRELATION OF EXPERIMENTAL DATA FOR $0 \leq \theta \leq 105$ DEG

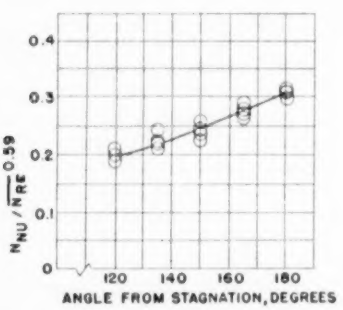

FIG. 5 CORRELATION OF EXPERIMENTAL DATA FOR 120 DEG $\leq \theta \leq$ 180 DEG

For $\theta \geq 120$ deg, the following relation applies

$$N_{Nu} \alpha (N_{Re})^{0.59} \qquad [10]$$

and Fig. 5 shows the parameter $N_{Nu}/(N_{Re})^{0.59}$ plotted against the angle from stagnation for 120 deg $\leq \theta \leq$ 180 deg.

Reference to Fig. 3 suggests that for 105 deg $\leq \theta \leq$ 120 deg, a satisfactory method of calculating the value of the local coefficient is to interpolate linearly between θ_{105} and θ_{120}.

In order to compare the coefficient with published data for spheres (9), a plot of the local coefficient was made on polar coordinates, and integrated over the surface using the theorem of Pappus. Fig. 6 shows the correlation obtained. The average coefficient for the sphere may be represented by

$$N_{Nu} = 0.37 (N_{Re})^{0.53} \qquad [11]$$

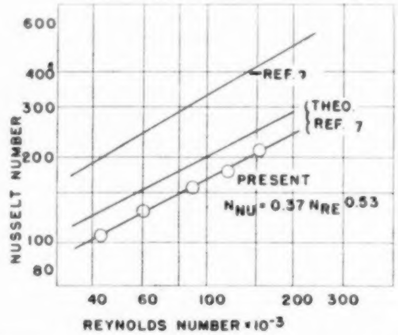

FIG. 6 CORRELATION OF EXPERIMENTAL DATA FOR A SPHERE

Comparison to the relation quoted in reference (9), namely

$$N_{Nu} = 0.33 (N_{Re})^{0.60} \qquad [12]$$

shows that the results obtained in the present investigation are about 50 per cent lower than those calculated from reference (9).

In an effort to examine this discrepancy, the original data of three of the studies from which the correlation of reference (9) is derived were considered. In the work of Buttner (10), it is not clear how the temperature of the sphere was measured in forced-convection studies. In the case of free convection, a wire probe was employed to determine the temperature gradient in the boundary layer. There was apparently no effort to minimize the turbulence of the air. Indeed, this turbulence was considered the more natural, since Buttner was concerned with ventilation, and the air issuing from a ventilator is probably turbulent. The use of a hand anemometer for measuring air velocity may have been unsatisfactory.

Borne (11) obtained heat-transfer coefficients from the evaporation of water through a porous-walled porcelain sphere, which seems to be subject to more inherent errors than the present investigation.

Johnstone, et al. (7), measured coefficients by allowing small particles of a known initial temperature to fall through a tubular furnace the walls of which were maintained at a constant temperature throughout the length of the furnace. The temperature rise of the particles was determined by collecting the particles in a calorimeter, and measuring the temperature rise of the water. The heat-transfer coefficient was computed using the log-mean-temperature difference. Reynolds numbers were limited to less than 100.

As a further means for comparison, the behavior of the coefficient at the stagnation point was considered. If it be agreed that the coefficient along the surface of the body may be expressed in terms of the local velocity and the distance of the point in question from the stagnation point, that is

$$h \alpha \sqrt{Vx} \qquad [13]$$

as quoted by Goldstein (12), then since for cylinders and spheres, respectively, according to potential flow

$$V_C = 2\ V_0 \sin \theta$$

$$V_S = (3/2)\ V_0 \sin \theta$$

the coefficient for a sphere should be smaller at every point where the boundary layer is laminar than that for a cylinder. In Fig. 7 the value of the Nusselt number at the stagnation point is compared with the values obtained for a cylinder by Schmidt and Wenner (1), with the theoretical value for a sphere computed from Tomotika's work, and with the values computed from reference (9). It will be seen that the coefficients for the sphere are in fact lower than those for the cylinder, and the agreement with theoretical values is quite good.

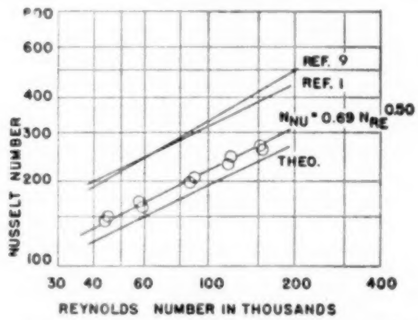

Fig. 7 Correlation of Experimental Data at Stagnation Point

On the other hand, the data of reference (9) for spheres are shown to be approximately the same as the values of Schmidt and Wenner for the stagnation-point coefficient for a cylinder. It should be noted that, since the value of the coefficient at stagnation is considerably higher than the average coefficient over the whole surface, a stagnation-point coefficient obtained from the data of reference (9) would be higher than that for a cylinder. This is probably the strongest argument for applying the data of reference (9) with reservations.

Finally, in Fig. 6 the average coefficient for the sphere is compared with the theoretical value of reference (7). It will be seen that the coefficients obtained in the present study lie approximately 18 per cent lower than theoretical values. It should be remembered, however, that Jakob expressed some doubt about the validity of the work of Boussinesq from which the theoretical expression of reference (7) was taken.

In conclusion, therefore, it may be said that the results of the present investigation, although yielding values lower than those which might have been deduced from existing data, appear to be somewhat more compatible with other facts.

Acknowledgment

Appreciation for the interest of Mr. W. F. Walker, Air Conditioning Engineer, Douglas Aircraft Company, is expressed. Thanks are also due to Mr. R. W. Wheeler, and to Dr. A. L. Klein, and Mr. Harry H. Hauger who read the manuscript. The author is grateful to the Douglas Aircraft Company for making this investigation possible.

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An Investigation of Heat Transfer From an Inclined Flat Plate in Free Convection[1]

By B. R. Rich,[2] Burbank, Calif.

The differential equations for heat transfer from vertical plates in free convection were rederived in dimensionless form and extended to include inclined plates. The analysis indicated that the only change in the plate was the component of the gravitational force in the plate surface direction. The limitation of the equation is that the flow be laminar and two-dimensional. Heat transfer from a flat plate in free convection was studied at angles of inclination ranging from 0 to 40 deg measured from the vertical, for a range of Grashof numbers from 10^6 to 10^9 and temperature differences from the surface to the surroundings of 200 to 260 F. The investigation shows the inclined-plate nondimensional unit thermal conductance, Nu, can be predicted from vertical plate conductances. The Grashof number in the Nusselt relation for the vertical plate, when modified by the cosine of the angle of inclination, can predict the inclined-plate conductances to within 10 per cent of those determined experimentally.

Nomenclature

The following nomenclature is used in the paper:

A = area of heat-transfer surface, sq ft
a = thermal diffusivity, sq ft/hr = $k/\gamma\ C_p$
c_p = specific heat, Btu/lb deg F
E = voltage across the resistance heating element, volts
g = gravitational constant, ft/sec^2
h_{cx} = local convective unit thermal conductance at plate surface, Btu/hr sq ft deg F
$h_{c\,avg}$ = average convective unit thermal conductance at plate surface, Btu/hr sq ft deg F
h_r = unit thermal conductance for radiation, Btu/hr sq ft deg F
k_s = thermal conductivity of air at surface temperature of plate, Btu/hr sq ft deg F/ft
l = length of the plate, ft
L_0 = length of path of light considered (width of plate, in.)
L = width of plate corrected for end effects, inches = L_0 + $0.05\ L_0$
ΔN = fringe shift used in calculation of density, dimensionless
q_c = rate of heat loss by convection, Btu per hr
q_{cond} = rate of heat loss by conduction, Btu per hr
q_r = rate of heat loss by radiation, Btu per hr
R_e = electrical resistance of heating element, ohms
T_0 = temperature of ambient air, deg R or deg F
T_s = temperature of the plate surface, deg R or deg F
T = temperature, deg R or deg F
u = velocity component in the x-direction, fps
v = velocity component in the y-direction, fps
x = co-ordinate along length of plate, in.
y = co-ordinate normal to the plate surface, in.
β = coefficient of expansion = $1/T_0$, deg R^{-1}
γ = weight density of air, lb per cu ft
ρ = mass density of air, slugs per cu ft
λ = wave length of light, microns
μ = absolute viscosity of air, lb sec/ft^2
ν = kinematic viscosity of air = μ/ρ = sq ft/sec
θ = angle of inclination of plate from the vertical, deg
ϕ = angle of inclination of fringe from the normal, deg
σ = Stefan-Boltzmann radiation constant
Gr_x = $g\beta\Delta T x^3/\nu^2$ = local Grashof number, dimensionless
Nu_x = $h_{cx}\, x/k_s$ = local Nusselt number, dimensionless
Pr = $3600\, g\mu\, C_p/k_s$ = Prandtl number, dimensionless
ξ = dimensionless length co-ordinate = $\sqrt[4]{\dfrac{\mathrm{Gr}}{4}}\dfrac{y}{x}$
φ = dimensionless temperature coefficient = $\dfrac{T - T_0}{T_s - T_0}$
$\propto$ = proportionality sign

Introduction

The mechanism of free convection from a vertical flat plate has been thoroughly investigated during the past 30 years. The solution of the equation for a laminar viscous fluid was first presented by Blasius and was extended in 1921 by E. Pohlhausen to include heat transfer. Since then numerous investigators, such as E. Schmidt and W. Beckmann, have corroborated the Pohlhausen solution experimentally.

The method of dimensional analysis has long been used by investigators to correlate results of experimental studies. In 1915 Nusselt made the first presentation of heat-transfer data in nondimensional form with the free-convection equation

$$\mathrm{Nu} = f\,(\mathrm{Gr}\cdot\mathrm{Pr})$$

where the dimensionless numbers are based on surface conditions. Data presented in this manner can be applied to other fluids and conditions as well as correlate data of other investigators.

A search of the literature reveals that where data were plentiful for free convection from vertical flat plates in laminar and turbulent flow, no data, analytical or experimental, could be found for inclined plates.

The differential equations for heat transfer from vertical plates in free convection were re-examined in nondimensional form and were extended to include inclined plates. The analysis indicated that the only change in the differential equations of the inclined plate with those of the vertical plate was the component of the gravitational force in the plate-surface direction. The limitation of the derived equation is that the flow be laminar and two-dimensional.

An experimental investigation using a Mach-Zehnder interferometer was undertaken to determine the local unit thermal conductances of vertical and inclined plates in free convection, and to corroborate the analytical analysis that vertical plate conductances could be modified to predict inclined plate conductances.

The interferometer data were analyzed using the fringe-shift method of interferogram evaluation. The equations used in de-

[1] Based on the author's Master of Science thesis at the University of California at Los Angeles.

[2] Aerodynamicist, Lockheed Aircraft Corporation.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, May 27, 1952. Paper No. 52—F-20.

termining the surface temperature and the temperature gradient are presented in the Appendix. The derivations of these equations and the operation of an interferometer may be found in references (1, 2)[2] or any other paper on interferometry. Owing to the fact that the mirrors in the interferometer were of inferior quality, the fringe pattern was initially curved; therefore the temperature-gradient equation had to be rederived to incorporate initial curvature and is presented in the Appendix.

Theoretical Analysis

Consider the differential equations for heat transfer from a vertical flat plate, Fig. 1.

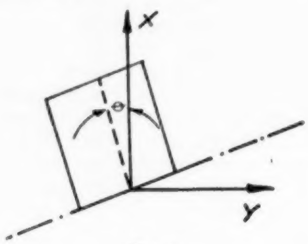


Fig. 1

Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad [1]$$

Momentum equation

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = \nu\frac{\partial^2 u}{\partial y^2} + g\beta\,(T - T_0) \qquad [2]$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = \nu\frac{\partial^2 v}{\partial y^2} \qquad [2']$$

Energy equation

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = a\frac{\partial^2 T}{\partial y^2} \qquad [3]$$

When the flat plate is inclined θ from the vertical, Fig. 1, Equations [1], [2], [2′], and [3] then become

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad [1a]$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = \nu\frac{\partial^2 u}{\partial y^2} + g\cos\theta\,\beta(T - T_0) \qquad [2a]$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = \nu\frac{\partial^2 v}{\partial y^2} + g\sin\theta\beta\,(T - T_0) \qquad [2a']$$

(since $v <<< u$, Equation [2a′] can be neglected)

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = a\frac{\partial^2 T}{\partial y^2} \qquad [3a]$$

Redefine the following dimensionless groups

$$T^* = \frac{T - T_0}{T_s - T_0} = \frac{T - T_0}{\Delta T}$$

where

$x^* = \dfrac{x}{l}$ $\quad T_0$ = ambient temperature

T_s = surface temperature

$y^* = \dfrac{y}{l}$ $\quad l$ = length of plate

a = thermal diffusivity

$$u^* = \frac{ul}{a}$$

$$v^* = \frac{vl}{a}$$

Substituting the dimensionless groups in Equations [1a], [2a], [3a]

$$\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} = 0 \qquad [4]$$

$$u^*\frac{a^2}{l^3}\frac{\partial u^*}{\partial x^*} + v^*\frac{a^2}{l^3}\frac{\partial u^*}{\partial y^*} = \nu\frac{a}{l^3}\frac{\partial^2 u^*}{\partial y^{*2}} + g\cos\theta\beta T^*\Delta T \qquad [5]$$

$$u^*\frac{\partial u^*}{\partial x^*} + v^*\frac{\partial v^*}{\partial y^*} = \frac{\nu}{a}\frac{\partial^2 u^*}{\partial y^{*2}} + \frac{l^3 g\beta\Delta T}{a^2}(\cos\theta)T^* \qquad [5a]$$

$$u^*\frac{\partial u^*}{\partial x^*} + v^*\frac{\partial v^*}{\partial y^*} = \mathrm{Pr}\frac{\partial^2 u^*}{\partial y^{*2}} + \mathrm{Pr}^2\mathrm{Gr}(\cos\theta)T^* \qquad [5b]$$

$$u^*\frac{a}{l^2}\frac{\partial T^*}{\partial x^*} + v^*\frac{a}{l^2}\frac{\partial T^*}{\partial y^*} = \frac{a}{l^2}\frac{\partial^2 T^*}{\partial y^{*2}} \qquad [6]$$

$$u^*\frac{\partial T^*}{\partial x^*} + v^*\frac{\partial T^*}{\partial y^*} = \frac{\partial^2 T^*}{\partial y^{*2}} \qquad [6a]$$

where

$$\mathrm{Pr} = \frac{\nu}{a}$$

$$\mathrm{Gr} = \frac{l^3 g\beta\Delta T}{\nu^2}$$

Writing a heat-balance equation for a point x at the plate surface

$$h\Delta T = -k\left(\frac{dT}{dy}\right)_{y=0} \qquad [7]$$

$$\frac{h}{k} = -\left(\frac{dT}{dy}\right)_{y=0}\frac{1}{\Delta T} \qquad [7a]$$

Multiply both sides of Equation [7a] by x

$$\mathrm{Nu}_x = \frac{hx}{k} = \frac{-x}{\Delta T}\left(\frac{\partial T}{\partial y}\right)_{y=0} \qquad [8]$$

But

$$\left(\frac{dT}{dy}\right)_{y=0} \alpha \left(\frac{dT^*}{dy^*}\right)_{y^*=0}$$

therefore

$$\mathrm{Nu}_x\ \alpha \left(\frac{dT^*}{dy^*}\right)_{y^*=0} = f(\mathrm{Pr}, \mathrm{Gr}\cos\theta, x^*, y^*) \qquad [9]$$

From the original differential-equation solution for a vertical plate (3)

$$\mathrm{Nu} = f(\mathrm{Pr}, \mathrm{Gr})$$

From the foregoing when the vertical plate is inclined

$$\mathrm{Nu} = f(\mathrm{Pr}, \mathrm{Gr}\cos\theta)$$

Experimental Apparatus. A $1^1/_2$-in. × $2^1/_2$-in. ellipsoidal shape mirrored Mach-Zehnder interferometer was used in this investigation to determine the unit thermal conductance, Figs. 2 and

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

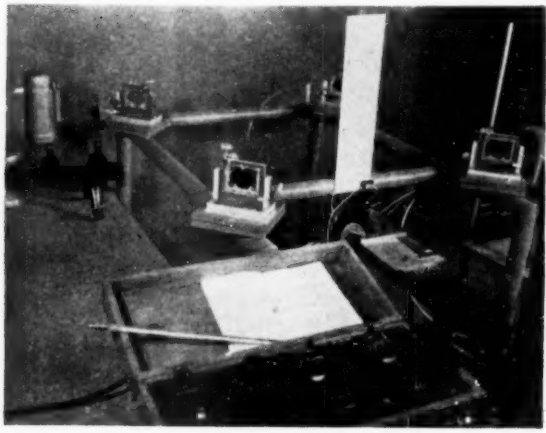


FIG. 2 THE MACH-ZEHNDER INTERFEROMETER

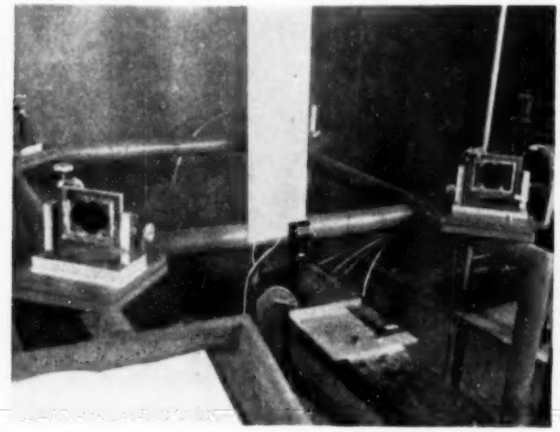


FIG. 3 FLAT PLATE POSITIONED FOR TEST IN INTERFEROMETER

3. A discussion on the interferometer[4] and the fringe-shift method of interferogram evaluation is presented in references (1) and (2).

An aluminum flat plate, 4 in. × 16 in. × $^{1}/_{8}$ in., was used as the test plate, Fig. 3. The plate was heated through the use of an Electrofilm resistance heating element applied on one side of the plate, the other side being the one to be investigated. The heating element was approximately 0.050 in. thick, with an over-all electrical resistance of 31.5 ohms.

On the side of the plate to be investigated, four chromel-alumel thermocouples (No. 40 B & S wires) were carefully spot-welded at 3, 6, 10, and 13 in., measured from the bottom of the plate, respectively.

The plate was mounted in a rotating vise, clamped securely at the bottom. Two pieces of micarta and asbestos were placed between the vise jaws and the plate to minimize the conduction loss. The vise was then clamped to the arbor of a drill-press stand so that vertical movement could be attained.

A 5-amp Variac was placed in the electrical circuit to regulate the voltage input, as the Electrofilm element has a temperature limit of 400 F. No constant-voltage transformer was used, as the data were taken in the evening when the voltage load was constant. Frequent checks were made on the voltage to confirm this.

The room temperature was recorded by means of a shielded thermometer suspended in the center of the room.

A thermopile radiometer was used to measure the radiation loss from the flat plate.

The test was run in an air-conditioned room, so that ambient temperature could be maintained relatively constant.

TEST PROCEDURE

The flat plate was positioned in one arm of the interferometer, Fig. 2. The interferometer was adjusted so that the plane of maximum interference coincided with the plane of the portion of the plate to be investigated.

The fringes then were oriented perpendicular to the plate surface. A tare interferogram was made of the unheated plate. The current was then turned on and voltage set at 78 volts. This voltage was chosen as it would heat the plate to a temperature of approximately 300 F. Temperatures were checked every 3 min until steady state was reached.

A photograph was then taken of the heated plate. Room temperature was recorded from a shielded thermometer suspended in the room. Plate-surface temperatures were recorded on a Leeds & Northrup potentiometer. Extreme care was taken throughout the procedure so that no air disturbances would be set up in the room. Input voltage and plate electrical resistance were measured. A radiometer reading was taken for every angular setting. The current was then shut off and the plate allowed to cool to room temperature. The plate was then reset and procedure repeated.

When the plate was inclined, an inclinometer was used to set the plate at the desired angle and the foregoing procedure followed.

Four points on the plate, 3, 6, 10, and 13 in. measured from the bottom of the plate, were investigated vertically and at four angles of inclination, 10, 20, 30, and 40 deg, respectively, measured from the vertical. As the data for the 40-deg inclined plate began to show the effects of the velocity component in the third dimension, larger angles of inclination were not studied as the interferometer is limited to two-dimensional problems. No attempt was made to determine the critical angle of inclination in this investigation.

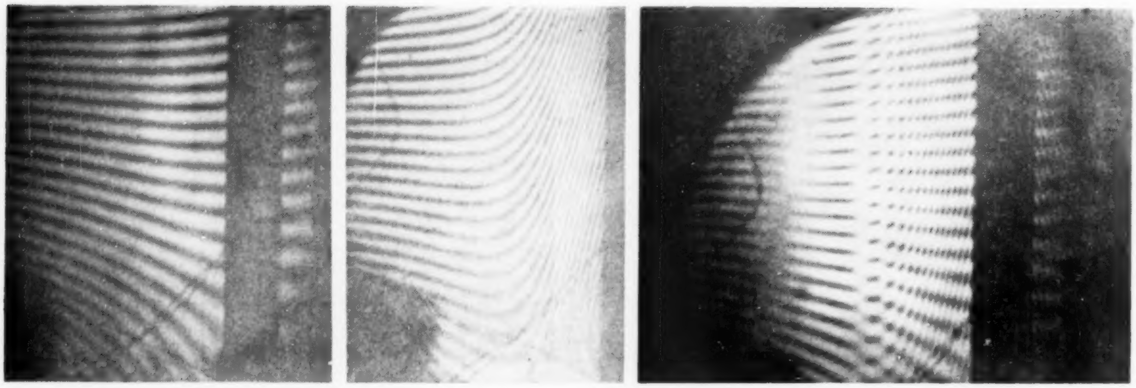
The plate was tested for a range of Grashof numbers from 10^{6} to 10^{9}, and for temperature differences from the surface and the surroundings of 200 to 260 deg F.

DISCUSSION AND EVALUATION OF EXPERIMENTAL DATA

Effect of Isothermal Fringe Curvature. Finite fringe interferograms were taken of the unheated and heated plate. Owing to the fact that the mirrors in the interferometer were of inferior quality, the fringes in the tare photograph (unheated plate) were initially curved; therefore the usual evaluation techniques could not be used directly. A physical method of correction as suggested by T. Zobel (7) was tried but failed to improve the fringe pattern appreciably.

The curvature would not affect the fringe count, which is used in determining the surface temperature as well as the temperature distribution in the boundary layer, since the curved fringes in the tare photograph act as datum. However, the calculation of the local temperature gradient at the plate surface would be in considerable error, since the temperature gradient at the plate surface is a function of the slope of the fringes in the heated-plate interferogram if the fringes in the tare photograph (unheated plate) are initially normal to the plate surface (1, 2). It was necessary to rederive an equation for the temperature gradient, which would incorporate the initial fringe curvature. The derivation of the modified temperature-gradient equation is presented in the Appendix.

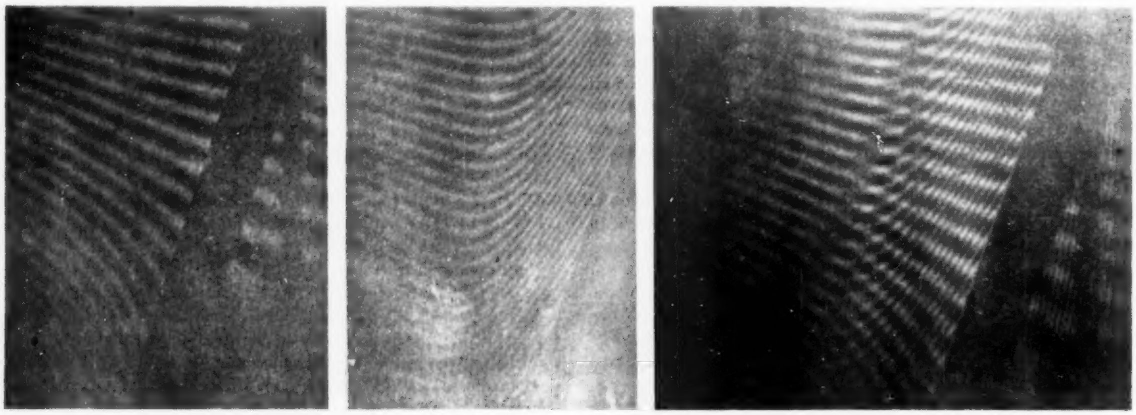
[4] The term "interferometer" in this paper will refer to the Mach-Zehnder interferometer.



(*a*) Unheated plate interferogram (*b*) Heated plate interferogram (*c*) Superimposed interferograms

Fig. 4 Vertical Plate in Free Convection

(Interferograms for point 3 in. measured from bottom of plate.)



(*a*) Unheated plate interferogram (*b*) Heated plate interferogram (*c*) Superimposed interferometer

Fig. 5 20 Deg Inclined Plate in Free Convection

(Interferograms for point 6 in. measured from bottom of plate.)

Interferogram Analysis. The procedure in analyzing the interferograms will be discussed briefly. Consider an unheated and heated-plate interferogram, Figs. 4 and 5. The two interferograms are superimposed and the unheated-plate or tare interferogram will act as a datum. The interferograms were analyzed in a Bausch and Lomb optical comparator with a magnification of 31.25; any further magnification was found to be impractical as the fringe sharpness decreased rapidly. The fringe shift, ΔN, is obtained by picking a fringe on the tare interferogram, using it as a datum, and counting the number of fringes on the tare interferogram that are intersected by the chosen fringe on the heated-plate interferogram from the point where the two fringes coincide in the free stream to where the heated fringe intersects the plate surface. The following other measurements are made:

ϕ = angle that isothermal fringe makes with normal to plate surface, deg

$\frac{\Delta x}{\Delta N}$ = distance between fringes measured along plate on-isothermal interferogram, in. per fringe

$\frac{\Delta y}{\Delta N}$ = distance between fringes measured normal to plate surface on heated interferogram, in. per fringe

θ = angle of inclination of plate, deg

t = plate thickness, in.

The plate thickness is measured to obtain the magnification factor of the interferogram resulting from the optical system of the interferometer, adjusting all readings accordingly.

Accuracy of Measurement. Five readings were made for each measurement and averaged so that a high degree of accuracy could be maintained. The over-all readings ranged from 0.012 to 0.150 in., with a maximum variation between any two readings of a measurement of 0.001 in. The least accurate of all the measurements was the slope of the isothermal fringe which varied by as much as 25 per cent, but since the angle was to be multiplied by a small quantity, $\Delta N/\Delta x$, in the temperature-gradient equation, the error could be tolerated.

Effects of Aberration. No attempt was made in this investigation to correct for the effects of aberration. It was shown in reference (8) that the error, due to aberration resulting from imperfect collimation, would be negligible for most investigations.

End Effects. The derivation of the equations for the fringe-shift method (1, 2) were based on the distance that light passes through the heated region. Owing to the fact that the heated region extends beyond the width of the plate, it is necessary to use

a factor to correct for the end effects. R. Kennard (2) found that a factor of 5 per cent added to the width of his plate would compensate for the end effects. By coincidence this author's plate had the same width as Kennard's plate, 100 mm ($\cong$4 in.); therefore the same factor was used. The choice of this factor is purely empirical. This factor was later found to be too small as this investigation was conducted at higher surface temperatures, but the discrepancy did not warrant the recalculation of all the results.

Effects of Radiation and Conduction. In free-convection studies on heated bodies the effects of radiation losses are usually quite high. It was first presumed that if the plate were painted black with an emissivity close to unity, the radiation losses could be predicted. This method was good but had to be abandoned as the radiation loss was 80 per cent of the total heat loss, whereby a small discrepancy in the radiation loss would result in a large-percentage error in the convective loss. The plate was then painted silver reducing the radiation loss to about 59 per cent of the total heat loss. The amount of radiation loss was measured accurately by means of a calibrated thermopile radiometer. The equivalent conductance for radiation, h_r, was found to be 1.92 Btu/hr sq ft deg F.

The conduction loss from the plate was neglected as the loss from the end of the plate was minimized by the use of micarta jaws in the rotating vise and covering the exposed areas of the vise with asbestos.

Extreme caution was observed with the heating element as the resistance has a tendency to drift, that is, increase with age and use. The resistance was checked frequently and found to vary about 10 per cent over the entire investigation.

A heat balance was made for every angular setting.

Discussion of Results

Plate-Surface Temperature. The plate-surface temperatures were determined by the interference fringe-shift method (1, 2) (see Appendix) for the four points on the plate surface at different angles of inclination. The interferometer temperatures were compared with the thermocouple measured temperatures, Table 1. The former values were higher for most of the points investigated, with the discrepancy ranging from 0.5 F to 7.0 F for surface temperatures ranging from 284 F to 331 F. A small portion of the discrepancy can be attributed to the thermocouple-fin effect causing a local reduction in surface temperature, with the major discrepancy due to the choice of the end correction factor. The choice of the 5 per cent correction factor, as explained previously, was recommended by Kennard (2). No attempt was made to determine the end effects in this investigation as this would be a separate problem in itself. The low interferometer values or high thermocouple readings can be explained only as experimental error. Although closer correlation has been achieved by other investigators using interferometers, the error never exceeded 3 per cent, which is small when considering the use of a small inexpensive interferometer.

TABLE 1 COMPARISON OF INTERFEROMETER TEMPERATURES WITH THERMOCOUPLE TEMPERATURES

θ, deg	x, in.	T_0, deg F	T_s, deg F	T_s,[a] meas, deg F	Gr, 10^6	hc_x, Btu/hr sq ft deg F	Nu_x	Nu_{mod}[b]
0	3	80.2	284.3	283.0	4.4	1.36	16.9	
	6	79.4	303.7	305.5	37.8	1.14	27.8	
	10	76.6	306.7	315.0	178.0	0.98	39.8	
	13	74.5	334.2	327.8	424.0	0.87	44.5	
10	3	76.0	284.9	287.0	4.5	1.21	15.1	16.8
	6	74.8	311.9	308.5	39.6	1.03	25.2	27.7
	10	71.5	314.8	317.6	189.0	1.02	41.3	39.7
	13	78.1	312.6	319.3	394.0	0.90	47.7	44.4
20	3	76.4	297.8	298.2	4.8	1.22	15.1	16.8
	6	75.1	314.2	318.2	39.7	1.03	25.1	27.4
	10	74.0	329.1	323.0	193.0	0.92	37.0	39.8
	13	79.5	325.5	326.0	397.0	0.89	47.0	43.9
30	3	71.8	305.3	296.6	1.2	0.99	14.4	16.0
	6	71.5	313.4	310.0	40.4	0.93	26.9	26.3
	10	74.5	334.6	328.8	194.0	0.99	46.3	37.7
	13	75.5	320.1	316.0	408.0	0.86	52.9	42.1
40	3	75.8	297.5	296.0	1.2	1.07	15.8	15.8
	6	75.8	312.9	315.0	39.5	0.96	27.8	26.0
	10	75.6	317.5	322.0	186.0	0.82	38.7	37.2
	13	75.0	322.2	317.0	412.0	0.99	61.3	41.6

[a] Thermocouple-measured temperatures.

[b] Grashof number for vertical plate has been modified by cos θ; i.e., $Nu_{mod} = Nu_{vert} (\cos\theta)^{1/4}$.

Boundary-Layer Temperature Distribution. The boundary-layer temperature profile is presented in Figs. 6 to 10 for the vertical and inclined plate. In order to correlate the four points investigated on the plate surface, the curves are presented in dimensionless co-ordinates. The ordinate φ is the ratio of the temperature difference between the temperature at a point in the boundary layer and the ambient or room temperature to the maximum temperature difference, that is, the difference between the plate-surface temperature and room temperature. This term is often referred to as the nondimensional temperature coefficient. The abscissa ξ is the nondimensional length co-ordinate which is a function of the local Grashof number and is equal to

$$\sqrt[4]{\frac{Gr_x}{4}}\,\frac{y}{x}$$

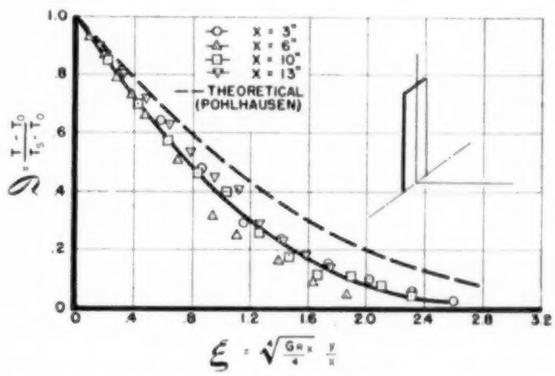


Fig. 6 Temperature Boundary-Layer Profile Along Heated Vertical Flat Plate

In Fig. 6 the boundary-layer temperature profile is plotted along with the Pohlhausen theoretical temperature distribution (3) for vertical plates in the laminar region. There is good correlation between the two curves close to the plate surface with the deviation increasing with increasing distance from the plate surface. The deviation results from the fact that the experimental data lie in the boundary-layer transition region between laminar and turbulent flow ($10^6 < Gr < 10^9$) which results in a steeper temperature gradient at the plate surface. According to E. Eckert (4), Gr equal to 10^9 is the critical Grashof number for turbulent flow.

As was mentioned previously, no data are available in the literature for inclined plates. To determine the effect of inclination on the local temperature distribution, the temperature profile for each point on the surface that was investigated was plotted for the different angles of inclination in Figs. 7 to 10. The data for the plate inclined 40 deg, particularly at the upper portion of the plate, Fig. 10, seem to be influenced either by the magnitude of the velocity component in the direction normal to the plate v, which in the analysis is neglected, now becoming significant or a third-dimensional effect, as the experimental points fluctuate about the other experimental points. Figs. 7 to 10 indicate that tilting the plate slightly decreases the slope of the gradient at the plate surface. The curves follow a general trend

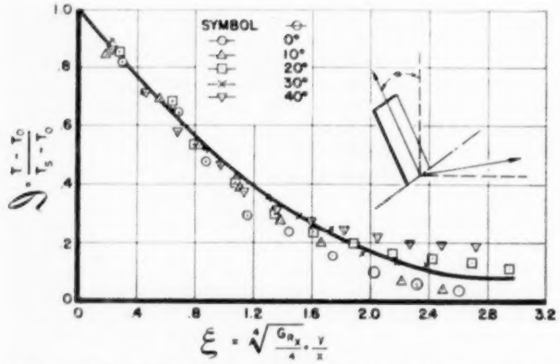


FIG. 7 TEMPERATURE BOUNDARY-LAYER PROFILE MEASURED 3 IN. FROM BOTTOM OF FLAT PLATE AT DIFFERENT ANGLES OF INCLINATION

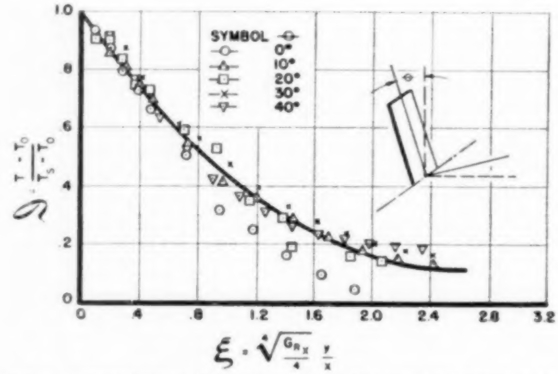


FIG. 8 TEMPERATURE BOUNDARY-LAYER PROFILE MEASURED 6 IN. FROM BOTTOM OF FLAT PLATE AT DIFFERENT ANGLES OF INCLINATION

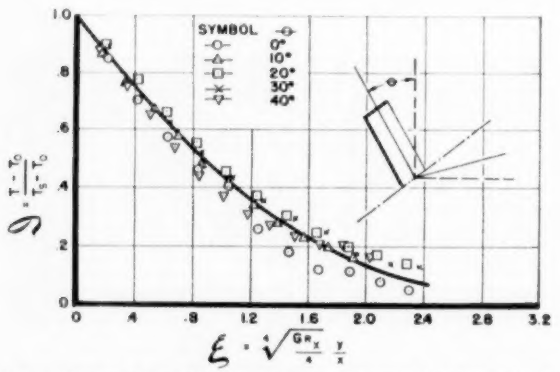


FIG. 9 TEMPERATURE BOUNDARY-LAYER PROFILE MEASURED 10 IN. FROM BOTTOM OF FLAT PLATE AT DIFFERENT ANGLES OF INCLINATION

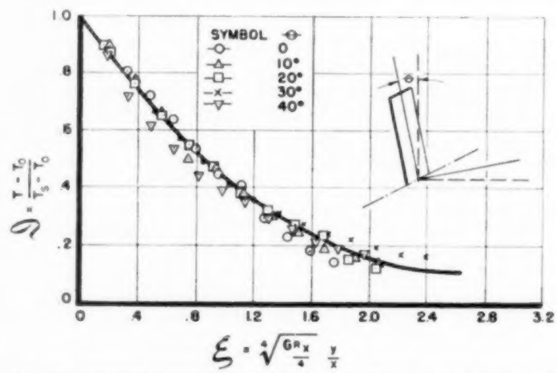


FIG. 10 TEMPERATURE BOUNDARY-LAYER PROFILE MEASURED 13 IN. FROM BOTTOM OF FLAT PLATE AT DIFFERENT ANGLES OF INCLINATION

of deviating from the distribution for the vertical plate with increasing distance from the plate surface with the greatest deviations at the bottom of the plate decreasing with increasing distance from the bottom of the plate. The point 13 in. from the plate surface, Fig. 10, exhibits excellent correlation for all angles of inclination.

Convective Unit Thermal Conductances. The convective unit thermal conductance is presented in the literature in dimensionless form as the Nusselt number. The Nusselt number, Nu, is usually given by the equation of the form

$$Nu = C(Gr\ Pr)^n$$

where C is a constant and n is an exponent generally equal to $^1/_4$ for laminar flow and $^1/_3$ for turbulent flow. The presentation of the equation in the foregoing form is a linear approximation of experimental data when plotted on log-log co-ordinates where n is the slope of the line and C is the intercept.

Unfortunately, most of the data presented in the literature are average values, thereby making it necessary to modify the experimental results which are local values in this investigation into average values before any comparisons can be made. The average unit thermal conductance is defined as follows

$$h_{avg} = \frac{\int_0^l h_x\, dx}{\int_0^l dx} = \frac{4}{3} h_x$$

where

h_{avg} = average unit thermal conductance
h_x = local unit thermal conductance
x = distance along plate, measured from bottom of plate

and the average Nusselt number, $Nu_{avg} = 4/3\ Nu_x$.

The local Nusselt experimental values for the vertical plates were modified using the 4/3 factor and are plotted in Fig. 11. Two equations for vertical surfaces, one from W. J. King's experiments (5) and the other from the studies made by Y. S. Touloukian and his coworkers (6), are also plotted in Fig. 11. The correlation between the results and King's equation is within 5 per cent whereas Touloukian's equation is 10 per cent higher than the experimental results. The experimental points indicate that if a curve were drawn it would not be a straight line, which indicates that a simple power function may not be sufficient.

The following equation by E. Eckert (4) for local conductances for vertical plates was the only local equation available in the literature

$$Nu_x = 0.508\ Pr^{1/2}\,(0.952 + Pr)^{-1/4}\,(Gr)^{1/4}$$

The Prandtl number, Pr, for air in the range of temperatures investigated (300 F) is equal to 0.684, reducing the Eckert equation to

$$Nu_x = 0.371\ Gr^{1/4}$$

where

Nu_x = local Nusselt number at point x
Gr_x = local Grashof number at point x
x = distance measured from bottom of plate

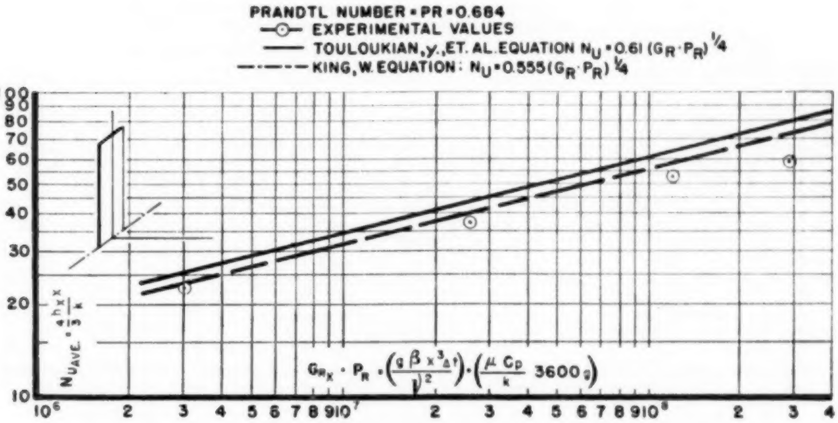


FIG. 11 AVERAGE NUSSELT NUMBER VERSUS GRASHOF-PRANDTL NUMBER
(Vertical flat plate in free convection.)

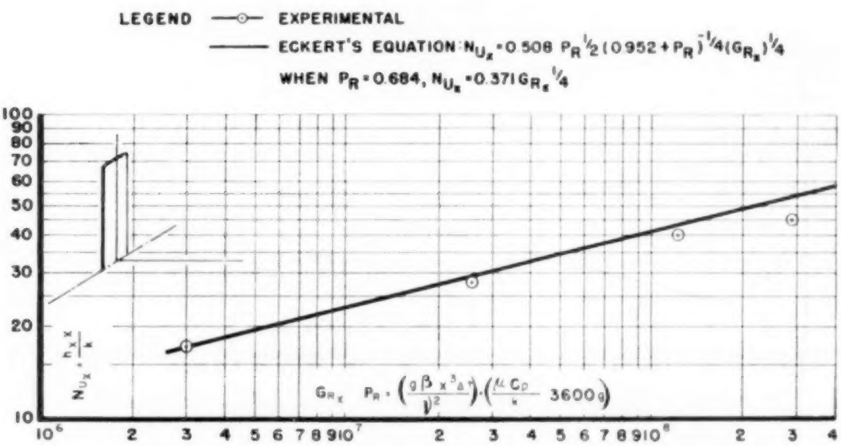


FIG. 12 LOCAL NUSSELT NUMBER VERSUS LOCAL GRASHOF-PRANDTL NUMBER
(Vertical flat plate in free convection.)

In Fig. 12 the reduced equation is plotted along with the experimental points for a vertical plate. The experimental point at Gr Pr $= 2.9 \times 10^8$ is believed in error and will not be considered in the discussion. The experimental point at Gr Pr equal to 3×10^6 coincides with Eckert's curve and varies not more than 5 per cent at any other point. This correlation is excellent as the equation is a linear approximation of Eckert's experimental data. The close correlation between Eckert's equation for local values justifies the existence of the local Grashof number for the range of Grashof numbers investigated ($10^6 < \mathrm{Gr} < 10^9$).

In order to see if the local vertical-plate conductances could be modified to predict inclined-plate conductances, the Grashof numbers for the vertical plate were modified by the cosine of the angle of inclination. This modification was predicted analytically in the re-examination of the differential equations. The experimental results for the vertical plate and the Eckert equation were modified and plotted along with the actual experimental points for the plate inclined at angles ranging from 10 to 40 deg in Figs. 13 to 16, respectively. The data definitely show that the inclined plate and conductances can be predicted to within 10 per cent by modifying the vertical-plate conductances. The limitation for this prediction is that the flow along the plate be laminar and two-dimensional. In Figs. 13 and 14, for the plate inclined 10 and 20 deg, respectively, the modified experimental vertical-plate conductances are 10 per cent higher than the actual inclined conductances. The per cent correlation improves as the inclination increases, showing perfect correlation at 40 deg. The data for the inclined plate show that the critical Grashof number for the turbulent flow is lowered by increasing the angle of inclination. The experimental point at the top end of the curve (i.e., at the top of the plate) for the 40-deg inclined plate, Fig. 16, definitely lies in the turbulent region, and therefore was not considered in the discussion.

Heat Balance. A heat balance was made for the vertical plate and for the plate at different angles of inclination. At equilibrium the electrical energy input should equal the heat loss by conduction, convection, and radiation. The heat loss by conduction was neglected as precautions were taken to minimize this loss. The calculated heat output for all positions agreed within 7 per cent of the total energy input.

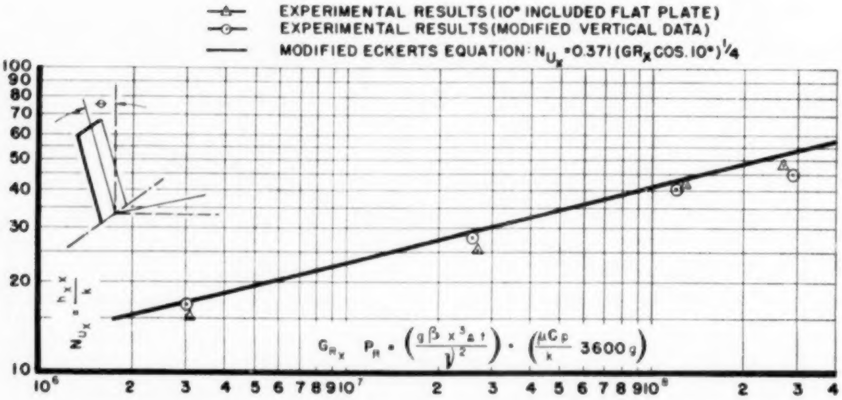


Fig. 13 Local Nusselt Number Versus Local Grashof-Prandtl Number
(Free convection from a heated flat plate inclined 10 deg from vertical.)

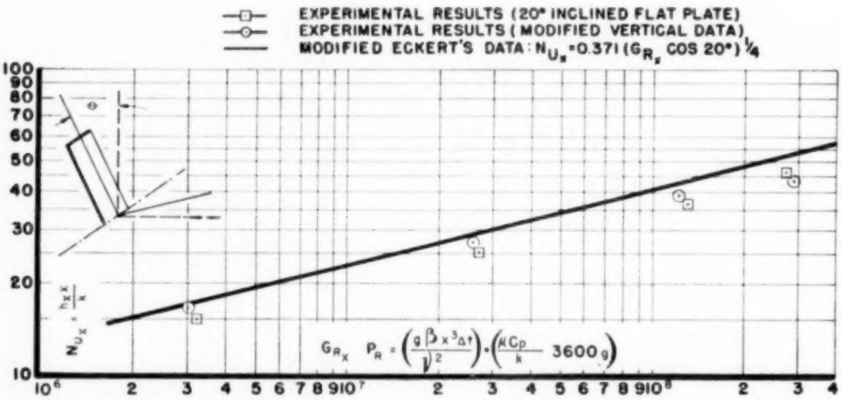


Fig. 14 Local Nusselt Number Versus Local Grashof-Prandtl Number
(Free convection from a heated flat plate inclined 20 deg from vertical.)

Conclusions

The following conclusions are drawn for the range of Grashof numbers investigated ($10^6 < Gr < 10^9$) in the study of heat transfer from vertical and inclined plates in free convection:

1 Inclined-plate unit thermal conductances can be predicted to within 10 per cent from modified vertical unit thermal conductances. The local nondimensional unit thermal-conductance equation for a vertical plate is modified by multiplying the local Grashof number by the cosine of the angle of plate inclination. (The modified Eckert equation: $Nu_x = 0.508\ Pr^{1/2}\ [0.952 + Pr]^{-1/4}\ (Gr_x \cos\theta)^{1/4}$, where θ is measured from the vertical.) The limitation of the prediction is that the flow be laminar and two-dimensional.

2 The close correlation between the data and the Eckert equation for local Nusselt and Grashof numbers justifies the existence of local values of the nondimensional groups.

Acknowledgments

The author wishes to acknowledge his indebtedness to Dr. Myron Tribus for his invaluable criticisms and patient guidance throughout this investigation. Appreciation is expressed to Dean L. M. K. Boelter and the Department of Engineering at the University of California at Los Angeles for the use of the laboratory facilities. Sincere thanks are extended to Mr. Paul Castenholz for his design and aid in assembling the interferometer.

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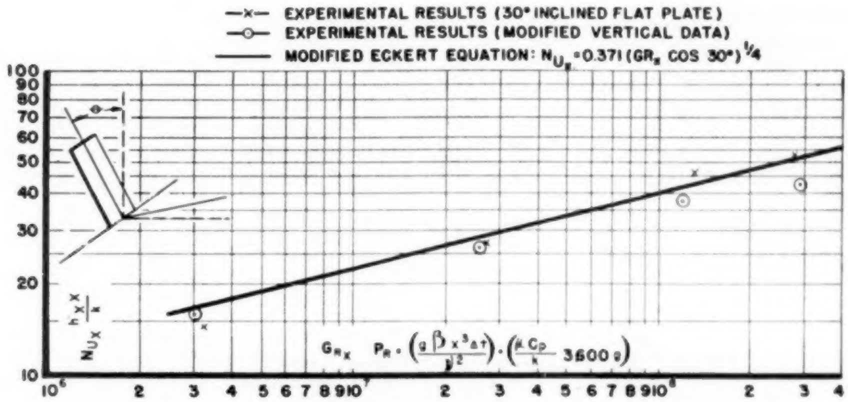


Fig. 15 Local Nusselt Number Versus Local Grashof-Prandtl Number
(Free convection from a heated flat plate inclined 30 deg from vertical.)

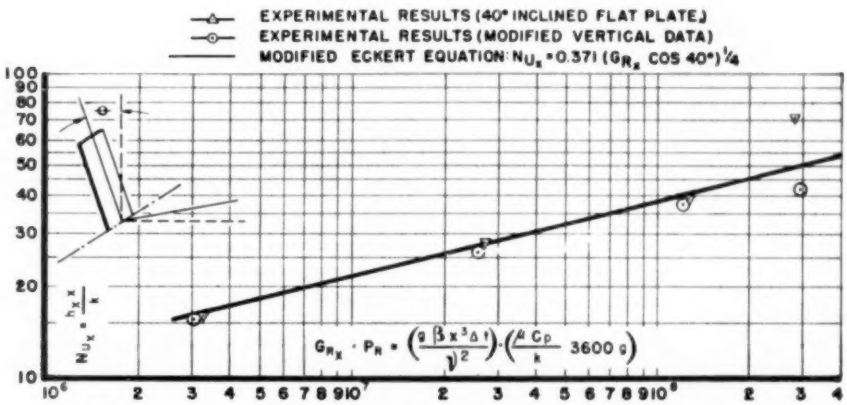


Fig. 16 Local Nusselt Number Versus Local Grashof-Prandtl Number
(Free convection from a heated flat plate inclined 40 deg from vertical.)

Appendix

Derivation of Temperature-Gradient Equation for Interferograms With Initially Curved Fringes

Consider a finite fringe interferogram number of fringes

$$N = f(x, y, T) \qquad [10]$$

Writing Equation [10] as a total differential

$$dN = \left.\frac{\partial N}{\partial x}\right]_{y,T} dx + \left.\frac{\partial N}{\partial y}\right]_{x,T} dy + \left.\frac{\partial N}{\partial T}\right]_{x,y} dT \qquad [11]$$

$$\frac{dN}{dy} = \left.\frac{\partial N}{\partial x}\right]_{y,T}\frac{dx}{dy} + \left.\frac{\partial N}{\partial y}\right]_{x,T} + \left.\frac{\partial N}{\partial T}\right]_{x,y}\frac{dT}{dy} \qquad [11a]$$

Rewriting Equation [11a] for dT/dy

$$\frac{dT}{dy} = \frac{\frac{dN}{dy} - \left.\frac{\partial N}{\partial y}\right]_{x,T} - \left.\frac{\partial N}{\partial x}\right]_{y,T}\frac{dx}{dy}}{\left.\frac{\partial N}{\partial T}\right]_{x,y}} \qquad [12]$$

and

$$\left(\frac{\partial N}{\partial y}\right)_T = \frac{\partial N}{\partial x}\frac{\partial x}{\partial y} + \frac{\partial N}{\partial T}\frac{\partial T}{\partial y} \qquad [13]$$

But

$$\frac{\partial T}{\partial y} = 0 \qquad [13a]$$

Therefore

$$\frac{\partial N}{\partial y} = \frac{\partial N}{\partial x}\frac{\partial x}{\partial y}$$

Referring to Fig. 17

$$\frac{\partial x}{\partial y} = \frac{\Delta x}{\Delta y} = \tan\phi \qquad [14]$$

Combining Equations [13a] and [14]

$$\frac{\partial N}{\partial y} = \frac{\partial N}{\partial x}\tan\phi \qquad [15]$$

Substituting in Equation [12]

$$\frac{dT}{dy} = \frac{\frac{dN}{dy} - 2\left.\frac{\partial N}{\partial x}\right]_{y,T}\tan\phi}{\frac{\partial N}{\partial T}} \qquad [16]$$

and from References 1, 2

$$\frac{\partial N}{\partial T} = \frac{(n-1)L}{\lambda_0 T_s} \quad [17]$$

$$\therefore \frac{dT}{dy} = \frac{\left.\frac{dN}{dy} - 2\frac{\partial N}{\partial x}\right]_{y,T} \tan\phi}{\frac{(n-1)L}{\lambda_0 T_s}} \quad [18]$$

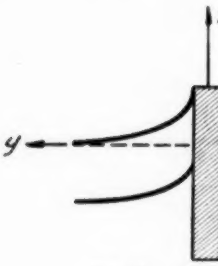


FIG. 17

Therefore, if fringes are perpendicular to plate, then $\Delta x/\Delta y = \tan\phi = 0$ and the equation reduces to the normal temperature-gradient equation for straight fringes

$$\frac{dT}{dy} = \frac{\frac{dN}{dy}}{\frac{(n-1)L}{\lambda_0 T_s}} \quad [19]$$

EQUATIONS USED IN ANALYSIS

1 Plate surface temperature, T_s

$$T_s - T_0 = \Delta T = \frac{\Delta N\ T_0}{\frac{(n-1)L}{\lambda_0} - \Delta N}, \text{ deg R or deg F}$$

(Equation derived in references 1 and 2)

2 Surface-temperature gradient, $\frac{dT}{dy}$

$$\frac{dT}{dy} = -\frac{\left.\frac{dN}{dy} - 2\frac{\partial N}{\partial x}\right]_{T,y} \tan\phi}{\frac{(n-1)L}{\lambda_0 T_s}}, \text{ deg F per in.}$$

3 Local convective unit thermal conductance, h_{c_x}

$$h_{c_x} = -\frac{k_s \frac{dT}{dy}}{\Delta T}, \text{ Btu/hr sq ft deg F}$$

4 Heat balance

Heat input = heat lost by convection + heat lost by radiation + heat lost by conduction

(Note that heat loss by conduction is negligible.)

(a) Heat input, $Q_{in} = \frac{E^2}{R_s} \times 3.41$, Btu per hr

(b) Loss by convection, Q_c

$$Q_c = h_{c_{avg}} A (T_s - T_0), \text{ Btu per hr}$$

(c) Loss by radiation, Q_R

$$G = K\,(\text{mv}) + F\sigma T_H^4$$

where

G = irradiation on radiometer, Btu/hr sq ft
K = radiometer constant = 5.2
(mv) = radiometer reading, millivolts
F = shape factor of radiometer = 0.034
T_H = radiometer housing temperature, deg R
σ = Stefan-Boltzmann constant = 0.173×10^{-8}

$$Q_R = \frac{GA}{F}, \text{ Btu per hr}$$

Discussion

SIMON OSTRACH.[5] In the subject paper the treatment of the pressure-gradient terms for the inclined plate was the same as for the vertical plate. However, the second equation of motion (or momentum equation) for the inclined plate should be

$$\frac{\partial P}{\partial Y} = \rho g \sin\theta \frac{(T - T_\infty)}{T_\infty}$$

(which is different from the $\partial P/\partial Y = 0$ for the vertical plate) and hence it cannot be argued, as for the vertical plate, that the pressure-gradient term in the first equation of motion vanishes because there is no pressure variation across the boundary layer.

A pressure variation across the boundary layer does, in fact, exist for the inclined plate and is given by the foregoing equation. It can be seen from this expression that this variation is of the same order as the driving or buoyancy term for angles of plate inclination θ, whereby $\sin\theta \approx \cos\theta$. However, the pressure-gradient term in the first equation of motion can be shown to be negligible on the basis of the boundary-layer assumptions alone. It then should be kept in mind that as the plate inclination exceeds 30 deg a large pressure variation normal to the plate exists. Just what effect, if any, this has on the actual results or whether this could in some way explain the deviation of the theoretical from the experimental results should be discussed.

J. RUTKOWSKI[6] AND M. TRIBUS.[7] In Equations [2] and [2′] the symbols are not consistent with Fig. 1 and the nomenclature of the paper. The second term on the right in Equation [2′] should be zero if the nomenclature is to be followed.

The statement that $v <<< u$ in Equation [2′] is no longer true when the plate is tilted. The term $(\sin\theta)^{1/4}$ then enters the correct version of Equation [2′]; for example, at a 30-deg inclination $(\sin 30)^{1/4} = 0.84$ and the terms cannot be dropped on the basis proposed by the author.

The fact that the inclination angle enters Equation [2] as $(\cos\theta)^{1/4}$ means that in the range of angles studied this term varies only from 1 to 0.935.

It would have been desirable for the author to include a plot of data showing dimensionless temperature versus angle of inclination with the author's variable ξ as a parameter. The figures given seem to show that the data for different angles fall on one curve.

It would have been extremely interesting if the author had made probes of the velocity boundary layer to see how the angle of inclination affects the velocity distribution and the transition from laminar to turbulent flow.

The fact that the author could get the accuracy shown in Table

[5] Aeronautical Research Scientist, National Advisory Committee for Aeronautics, Lewis Flight Propulsion Laboratory, Cleveland, Ohio.

[6] University of Michigan, Ann Arbor, Mich.

[7] University of Michigan. Jun. ASME.

1, despite the inferior plates in the inexpensive interferometer, is creditable. It seems, however, that the effects of inclining the plate are small and for this problem a repetition of the work using a better apparatus would be justified.

Author's Closure

The pressure gradient, $\partial P/\partial y$, discussed by Mr. Ostrach, is a point well taken. Unfortunately a pressure survey of the boundary layer was not made; therefore, no check could be made.

In reply to the comment of Messrs. Rutkowski and Tribus on the effect of the velocity component v, when the plate is inclined, the author concurs that the v component becomes significant especially at the top of the plate at the higher angles of inclination.

The author recommends that if further investigations were made, a boundary-layer survey illustrating the effect of pressure and velocity distribution with plate inclination be made.

Design of a Square-Root-Extracting, Force-Balance, Pneumatic Transmitter, Including Derivation of Formulas

By ARNOLD GOLDBERG,[1] CHICAGO, ILL.

This paper describes the design of a device for measuring fluid flow in a pipe. It deals with a particular type of flow-meter, one that measures the differential pressure developed across an orifice or other primary element and puts out an air pressure directly proportional to flow rate. Square root must be extracted in such an instrument. The square-root-extracting part of the device is of the shaped displacer and mercury type, in which the displacer does not move. Formulas are derived for calculating the shape.

Nomenclature

The following nomenclature is used in the paper:

- a_b = circular cross-sectional area of inside of bell, Fig. 4, sq in.
- a_0 = a_b at zero flow, Fig. 4, sq in.
- A_C = circular cross-sectional area of chamber, Fig. 4, sq in.
- A_D = circular cross-sectional area of outside of bell, Fig. 4, sq in.
- A_F = annular cross-sectional area of bell that is submerged in mercury at zero flow, Fig. 4, sq in.
- A_M = circular cross-sectional area of top of bell that is not exposed to P, Fig. 4, sq in.
- A_R = effective area of reaction diaphragm, Fig. 4, sq in.
- da_b = differential area on inside surface of bell, Fig. 4, sq in.
- g = acceleration due to gravity, ft per sec²
- H_0 = pressure on bottom of bell, area A_F, at zero flow, Fig. 4, in. Hg
- k = empirical constant or expression relating flow rate of fluid in pipe to differential pressure across orifice plate
- $K_1 = \dfrac{A_C}{A_R}$ = reaction ratio = arbitrary constant, selected by designer, dimensionless
- $K_2 = A_C - A_D$ = volumetric constant = arbitrary constant selected by designer, sq in.
- L = rise in level of mercury inside bell above level at zero flow, Fig. 4, in. Hg
- L' = mercury pressure on differential area, da_b, Fig. 4, in. Hg
- ΔP = differential pressure across orifice plate, Fig. 1, psf
- P = output pressure from transmitter at any flow rate, Fig. 2, in. Hg
- P_1 = P at 100 per cent flow, $q = 1$, in. Hg
- P_d = output pressure due to action of differential pressure, Fig. 3, in. Hg
- P_e = output pressure due to action of square-root extractor, Fig. 3, in. Hg
- P_Y, P_Z = output pressures at points Y and Z, Fig. 3, in. Hg
- q = fractional rate of flow $= \dfrac{\text{rate of flow}}{\text{max rate of flow}} = \dfrac{Q}{Q_1}$, dimensionless
- Q = rate of flow of fluid in pipe, cfs
- Q_1 = rate of flow at full range of meter, cfs
- S = drop in mercury level in chamber below level at zero flow, Fig. 4, in.
- X = origin of co-ordinate system shown in Fig. 3
- Y = point on curve P where P_e is a maximum, Fig. 3
- Z = point of intersection of curves $P = f(q)$ and $P_d = f(q^2)$, Fig. 3
- ρ = density of fluid inside square-root chamber, in this case mercury, Fig. 4, lb per cu in.

Introduction

A common method of measuring the flow of a fluid in a pipe is to place an obstruction in the pipe, such as an orifice. A pressure difference will develop across the orifice, and by measuring this pressure difference one can determine the flow rate. The relation between the rate of flow of the fluid and the differential pressure associated with it is assumed to be

$$Q = k\sqrt{2g\Delta P}$$

This assumption holds true over a wide range of Reynolds numbers provided that conditions permit neglecting the expansion of the fluid as it passes through the orifice.

Note that the flow rate is proportional to the square root of the differential pressure. Therefore an instrument graduated in flow units would have a "square-root" scale if no means were provided to alter it. Usually a linear scale is desired and so it is necessary to extract the square root of the differential pressure. Also, a linear output is often necessary for use with automatic controllers.

This paper describes an instrument which accomplishes such a purpose. The instrument puts out an air pressure which is proportional to the square root of an applied differential pressure. It will be referred to as the "flow transmitter." The mechanical functioning of the instrument will be described next.

Differential-Pressure Transmitter

The flow transmitter is made from an instrument called the differential transmitter with added parts to extract square root. The latter instrument sends out an air pressure which is linearly proportional to an applied pressure difference. The differential-pressure transmitter, shown diagrammatically in Fig. 1, will be described. The pressures from each side of the orifice in the pipe are applied to opposite sides of a slack diaphragm, L. The resultant force acts on weigh beam B which is pivoted at A. The higher pressure is introduced beneath the slack diaphragm L so that the moment about A due to the differential pressure is clock-

[1] Consulting Engineer, Goldberg Engineering Company. Mem. ASME.

Contributed by the Industrial Instruments and Regulators Division and presented at the Fall Meeting, Chicago, Ill., September 8–11, 1952, of The American Society of Mechanical Engineers.

 Manuscript received at ASME Headquarters, July 15, 1952. Paper No. 52—F-41.

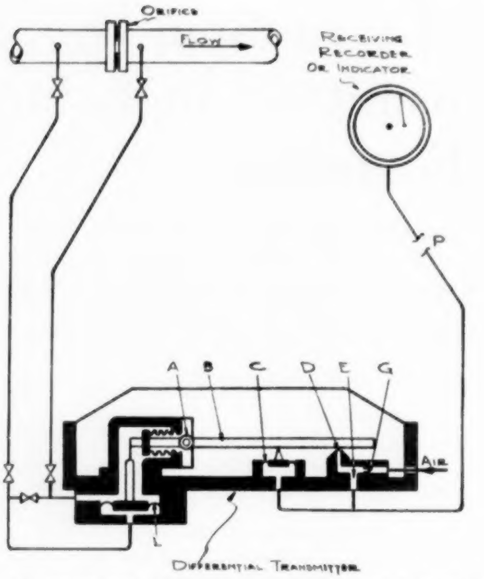


Fig. 1 Differential-Pressure Transmitter Shown Being Used as a Flowmeter

(Output air pressure is proportional to square of flow rate in pipe.)

wise. This moment is opposed by another slack diaphragm C which is called the "reaction diaphragm." The end of the weigh beam covers a bleed valve D which vents a chamber in the bleed block E. Air is introduced into the bleed block through an orifice G. The size of the orifice is such that its air capacity is always much larger than consumption and hence variations in air-supply pressure will not affect the accuracy of the instrument appreciably. The air pressure inside the bleed block is introduced under the reaction diaphragm C and is also the output pressure.

The function of the bleed nozzle and of the feedback of the output pressure to the reaction diaphragm is to maintain the weigh beam in a state of equilibrium. Any unbalanced force upward due to the differential pressure at L will cause the bleed valve D to close. The output pressure will then build up in the reaction diaphragm C until a state of equilibrium is again achieved.

Thus the instrument will always put out an air pressure which is linearly proportional to the applied differential pressure. The proportionality constant is determined by the ratio of areas of the two opposing diaphragms and by their leverages about point A.

Flow Transmitter

To convert the differential-pressure transmitter into the flow transmitter a bell and chamber are added, as shown in Fig. 2. The bell J is suspended from the weigh beam B and is immersed in a pool of mercury in the bottom of the chamber H. The inside of the bell is vented to atmosphere. The annular space between the bell and the chamber is sealed at the top by the reaction diaphragm C and becomes the reaction diaphragm chamber. As in the differential-pressure transmitter, the output pressure is fed back from the bleed chamber E to the reaction chamber H.

An unbalanced force upward at L will cause the output pressure to increase, the same as in the case of the differential transmitter just described. The increase in output pressure P will cause the mercury to descend in the annular space and to rise in the bell. As will be shown, the resultant of the forces acting on the bell, as the mercury rises inside it, is a force acting downward which aids the unbalance applied at L. The effect of the bell, then, is to produce a higher output pressure than would otherwise be obtained. At low flows the cross-sectional area of the bell is large, producing a large effect. At high flows the area is small and the additive effect is also small.

One might expect that the bell would exert an upward force on the beam as the mercury rose inside it. This is not the case, for the effect of the air pressure acting on top of the bell exceeds the effect of increased buoyancy, and the resultant force is down.

Square-root-extracting devices and particularly those that have feedback often suffer from stability problems. This design has a point in its favor, in that the change in output pressure at E due to an unbalance at L, Fig. 2, is felt almost instantaneously at C. The lag due to the inertia of the mercury does not delay this reaction.

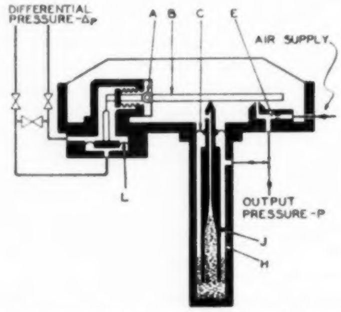
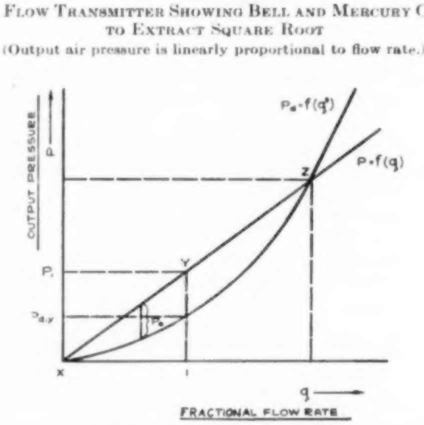


Fig. 2 Flow Transmitter Showing Bell and Mercury Chamber to Extract Square Root

(Output air pressure is linearly proportional to flow rate.)



$P = f(q)$. Transmitter output pressure versus fractional flow rate

$P_d = f(q^2)$. That part of output pressure developed by applied differential

$P_s = P - P_d$. That part of output pressure developed by square-root extractor

Fig. 3 Air Pressures Acting in Pneumatic Transmitter When Square Root Is Extracted

Discusssion of Pressures Acting in Square-Root-Extracting Transmitter

The two curves in Fig. 3 show output pressure plotted against fractional flow rate. Without a square-root-extracting device the transmitter puts out a pressure as shown by curve P_d. Since the transmitter is a linear instrument, the output is a square curve the same as the applied differential. It is desired that the transmitter have a characteristic depicted by curve P, a straight line. To accomplish this end, the square-root-extracting device

must add to the effect of the differential pressure below point Z and must subtract above point Z. The air pressure that is developed as a result of the action of the square-root extractor is called P_e and is equal to $P - P_d$.

At point Y the amount of pressure added is a maximum. Y may be a boundary point in some cases. If the device is an additive device only, and continues to add as the output pressure increases, then its operation must be confined between X and Y. Beyond Y, P_e must become less positive. The square-root-extracting bell which is dealt with in this paper is an additive device only and so operates between X and Y. In a recent thesis[2] devices are discussed which operate from X to Z, or from Y beyond Z, and so forth. The author is of the opinion that the device that operates between X and Y has the most merit.

Calculation of Shape of Bell

The following formulas are used to calculate the bell shape and are derived in the Appendix

$$K_1 = \frac{A_C}{A_R}, \qquad K_2 = A_C - A_D$$

$$S = \frac{P_1}{2K_1}(2q - q^2) \qquad [1]$$

$$L = q\,P_1 - S \qquad [2]$$

$$a_b = \frac{K_2(1 - q)}{K_1 - (1 - q)} \qquad [3]$$

$$a_0 = \frac{K_2}{K_1 - 1} \qquad [4]$$

To manufacture a bell shape a table of values is desirable showing the diameter at various locations. The diameter is found from a_b, Equation [3] and its location from L, Equations [1] and [2]. If P_1, A_C, A_D, and A_R are known, L and a_b may be computed for a series of chosen values of q.

The controller or recorder operates on a standard pressure. The other three constants in the equations are areas depicted in Fig. 4. They are selected by the designer and proportioned one against the other, bearing in mind the following factors: A_R should be selected large enough so that the forces developed will be large and therefore small amounts of friction will not impair the accuracy of the instrument. A_R is an effective area. Experiments show that this area is not the entire area of the slack diaphragm. It is a circular area whose diameter is the diameter of the diaphragm disk plus the width of the slack as shown in Fig. 4. This is approximately true when the diaphragm disk and the diaphragm plate are in line. As the diaphragm moves from such a position the effective area changes. The bell of this meter moves so little that the stated relation can be made to hold reasonably closely. From Equation [4] it can be seen that K_1 must always be larger than unity. Therefore A_C must always be larger than A_R. A_D must be sufficiently smaller than A_C so that the minimum clearance between the bell and the chamber wall is $^1/_4$ in. Otherwise, meniscus effects of the mercury will affect the operation of the instrument. Having selected these three areas, a_0 is determined as shown by Equation [4]. Check to see if a_0 is small enough to fit inside the bell.

Discussion of Equations

Consider Equation [1] and note an unusual circumstance. The distance the mercury falls, S, is independent of the bell shape. The values A_D and a_b do not occur in Equation [1]. Likewise the force the bell develops, given by Equation [12], is also independent of A_D or a_b. Therefore it can be deduced that an infinite number of bells can be made which will extract the square root for any given combination of P_1, A_C, and A_R. The question presents itself, which bell should be chosen?

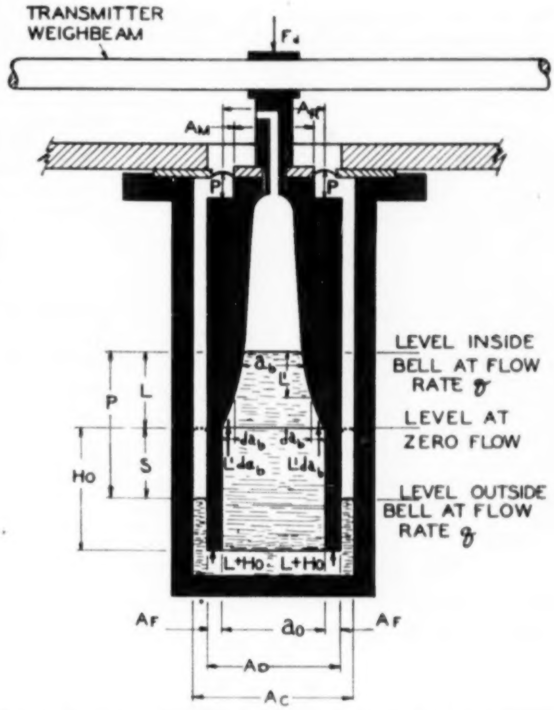


Fig. 4 Vertical Cross Section of Square-Root-Extracting Bell

(Horizontal cross section is circular.)

As discussed in the thesis referred to,[2] if a choice has been made for P_1 and A_R, there is only one bell which uses the least amount of mercury. This bell can be determined by trial and error using Equation [19] of the thesis[2] and different combinations of A_C and A_D.

However, the quantity of mercury required is not the most important factor. The speed of balancing is of importance as well as the stability of the instrument. The latter two factors are discussed at greater length in the thesis.

Summary

Most of the detail design of the transmitter is omitted in this paper. Rather, an attempt is made to study the fundamentals of extracting square root, the forces that act, and the regions to which these forces must be confined. The formulas used to calculate a shaped bell were presented and their application was discussed briefly. The derivation of these formulas is given in the Appendix.

This paper is not intended to describe any one design in detail. It is felt that it would be most useful if emphasis were placed on basic principles, and upon the derivation of the formulas. Hence it may serve as a guide to other designers in deriving new formulas for other related devices.

Acknowledgments

The author wishes to acknowledge the help of Mr. A. J. Rosenberger who originated much of the mechanical design, and the

[2] "Square Root Extracting Devices for Pneumatic Flow Meters," by Arnold Goldberg, thesis for master's degree, Illinois Institute of Technology, June, 1950.

help of Mr. J. B. McMahon in the preparation of the paper. The author also wishes to thank the Republic Flow Meters Company, Chicago, Ill., under whose auspices the foregoing work was done.

Appendix

Derivation of Equations

Referring to Fig. 3, it is shown that

$$P_{e.Y} = P_{d.Y} = \frac{P_Y}{2}$$

That is to say, when the corrective pressure is a maximum, it is equal to one half the total output pressure.

Y has been defined as that point where P_e is a maximum, that is

$$\frac{dP_e}{dq} = 0$$

Knowing that the curve P is a straight line through point P_Y, q_Y, its equation is therefore

$$P = \frac{P_Y}{q_Y} q \quad [5]$$

The equation of the second curve is a square curve through the point $P_{d.Y}$, q_Y and is given by

$$P_d = \frac{P_{d.Y}}{q_Y^2} q^2 \quad [6]$$

The difference between Equations [5] and [6] is

$$P_e = P - P_d = \frac{P_Y}{q_Y} q - \frac{P_{d.Y}}{q_Y^2} q^2 \quad [7]$$

Differentiating Equation [7]

$$\frac{dP_e}{dq} = \frac{P_Y}{q_Y} - 2q \frac{P_{d.Y}}{q_Y^2}$$

$$\left(\frac{dP_e}{dq}\right)_Y = 0 = \frac{P_Y}{q_Y} - 2q_Y \frac{P_{d.Y}}{q_Y^2}$$

$$P_Y = 2P_{d.Y} \quad [8]$$

Thus the maximum additive compensation that can be supplied by the square-root extractor is one half the output pressure at point Y.

P_Y is used as the maximum output pressure and so corresponds to P_1 defined in the nomenclature. It can be found in the nomenclature that by definition

$$q = \frac{Q}{Q_1}$$

$$q_Y = \frac{Q_1}{Q_1} = 1 \quad [9]$$

Thus the maximum additive compensation occurs at a fractional flow rate of one.

Substituting Equations [8] and [9] into [7]

$$P_e = \frac{P_1}{2} (2q - q^2) \quad [10]$$

A term is found in the following equivalent to P_e which converts Equation [10] into a useful equation.

The movement of the bell is of the order of magnitude of 0.003 in. or less through the range of the instrument and is neglected. The bell is in equilibrium and the forces acting upon it therefore add up to zero. These forces are depicted in Fig. 4. The downward direction is taken as the direction of positive force. Shown acting downward on the bell is the force F_d. It is produced by the differential pressure. If the bell were not present, the transmitter would have developed the reaction pressure P_d shown in Fig. 3. Therefore

$$F_d = P_d A_R$$

Using Equation [7]

$$F_d = P_d A_R = (P - P_e) A_R$$

Also acting downward is a force due to the air pressure on top of the bell

$$P(A_D - A_M)$$

The reaction diaphragm force acts upward upon the bell

$$P(A_R - A_M)$$

Two buoyancy forces act upward

$$(L + H_0) A_F + \int_{a_0}^{a_b} L' \, da_b$$

Writing Newton's first law for equilibrium

$$\Sigma F = (P - P_e) A_R + P(A_D - A_M) - P(A_R - A_M) - (L + H_0) A_F - \int_{a_0}^{a_b} L' \, da_b = 0$$

Some of these terms cancel. $H_0 A_F$ is removed mechanically in the instrument with counterweights. Solving explicitly for $P_e A_R$

$$P_e A_R = PA_D - LA_F - \int_{a_0}^{a_b} L' da_b \quad [11]$$

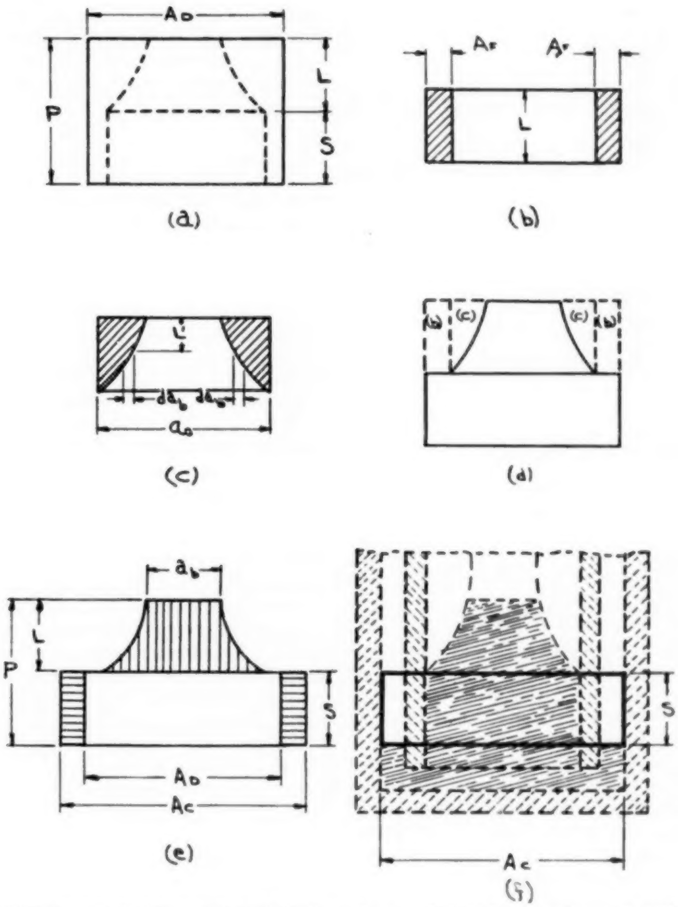
Fig. 5 is a graphical solution of Equation [11]. Each section of Fig. 5 is a part of the bell shown in Fig. 4. Each figure is a vertical cross section of a right circular volume. The volumes represented by these figures are equivalent to forces since their horizontal cross sections are areas and their altitudes are equal to the pressures which act on those areas. These sections of Fig. 5, then, represent forces that are acting on the bell. It will be shown that they are equivalent to the terms of Equation [11]. By adding and subtracting these volumes a solution can be found for Equation [11] in terms of the known constants of the bell.

Fig. 5 (*a*) shows a volume equal to the first term of Equation [11]. It is that section of the bell that lies between the two mercury levels as shown in Fig. 4. Fig. 5 (*b*) shows an annular cylindrical volume that is equivalent to the second term of Equation [11]. Its position in Fig. 4 can be determined by the dimension L. Fig. 5 (*c*) has a volume equivalent to the third term of Equation [11]. It represents the buoyancy force of the mercury acting on the shape. Fig. 5 (*d*) is the graphical sum of the three terms and shows the last two subtracted from the first.

Any mercury that rises up the center of the bell must have come from the annular volume outside the bell. In Fig. 5 (*e*), then, the volume shown crosshatched horizontally must be equal to the volume shown crosshatched vertically. Replacing the shaped volume by the annular volume in Fig. 5 (*d*), the cylinder shown heavy in Fig. 5 (*f*) is obtained. The volume of this cylinder is $A_C S$. Writing Equation [11] again and solving for P

$$P_e = \frac{A_C S}{A_R} \quad [12]$$

It seems unusual that the solution of such a complicated fo-



Figures are vertical cross sections of circular volumes. Vertical dimension is proportional to pressure and horizontal dimension is an indication of area upon which pressure acts. Volume of figures, then, is proportional to force.

(a) Volume = PA_D = first term of equation
(b) Volume = LA_F = second term of equation
(c) Volume = $\int_{a_0}^{a_b} L\, da_b$ = third term of equation
(d) Volume shown solid equals P_e
(e) Volume with horizontal section lines equals volume with vertical section lines
(f) Volume shown solid equals $P_e = A_cS$

FIG. 5 GRAPHICAL SOLUTION OF EQUATION $P_e = PA_D - LA_F - \int_{a_0}^{a_b} L' da_b = A_e S$

system should come out equal to such a simple value. In the thesis referred to,[2] solutions are given for other systems with equally simple results.

Going back to Equation [10], substitute Equation [12] for P_e and solve explicitly for S to obtain one of the basic equations

$$S = \frac{P_1 A_R}{2A_C}(2q - q^2) = \frac{P_1}{2K_1}(2q - q^2) \qquad [1]$$

Referring to Fig. 4, it can be seen that

$$P = L + S$$

$$P = qP_1 \qquad [13]$$

The last is by definition. Solving for L, Equation [2] is obtained

$$L = qP_1 - S \qquad [2]$$

As mentioned previously, any mercury that leaves the outer annular volume must show up inside the bell. Expressed in differential form

$$(A_C - A_D)\, dS = K_2\, dS = a_b\, dL$$

$$\frac{dS}{dL} = \frac{a_b}{K_2} \qquad [14]$$

Differentiating Equation [1] with respect to q

$$\frac{dS}{dq} = \frac{P_1}{2K_1}(2 - 2q) = \frac{P_1}{K_1}(1 - q) \qquad [15]$$

Substituting Equation [1] into [2] and differentiating

$$\frac{dL}{dq} = P_1 - \frac{P_1}{K_1}(1 - q) \qquad [16]$$

Dividing Equation [15] by [16]

$$\frac{ds}{dL} = \frac{\frac{P_1}{K_1}(1-q)}{\frac{P_1}{K_1}[K_1-(1-q)]} = \frac{(1-q)}{K_1-(1-q)} \quad [17]$$

Setting Equation [17] equal to [14] and solving for a_b, another of the basic equations results

$$a_b = \frac{K_2(1-q)}{K_1-(1-q)} \quad [3]$$

The initial area is the area at zero flow when $q = 0$

$$a_0 = \frac{K_2}{K_1-1} = \frac{A_C - A_D}{\frac{A_C}{A_R} - 1} \quad [4]$$

Discussion

J. A. Perry, Jr.[3] The design of an integral pneumatic square-rooting flow-measuring instrument meets many problems which at first may not be apparent.

In the field of flow measurement a seemingly infinite variety of devices has been developed to extract the square root of the differential impulse received from a primary element. Square-rooting devices probably can be divided into two general categories. The first is the integral device such as this, the Le Deaux bell, or the electrical-resistance-rod type. The other is the nonintegral or remote square-rooting type. This type receives the differential impulse from the measuring element and then by means of a cam or some other device extracts the square root.

The most obvious purpose for desiring a straight-line impulse is to be able to record flow on an evenly divided chart. Another reason is for totalizing flow. Obviously a totalizer must count at a rate proportional to the flow and not proportional to the differential head across an orifice.

Another reason for desiring a linear-flow impulse, and the major need for which this transmitter was designed, is for flow control. An ideal flow control would have a measuring impulse with an equal response throughout the range of operation. Mathematically, the output impulse of the square-root transmitter described by the author can be stated as, $I = CW$, where I is the pneumatic impulse and W is the flow. The change in impulse for a change in flow is

$$\frac{dI}{dW} = C$$

This means that at all rates of flow the transmitter output will change equally for equal changes in flow. However, the output of a conventional differential transmitter is $I = CW^2$. Here the change in impulse for a change in flow is

$$\frac{dI}{dW} = 2CW$$

Now, the impulse change is not constant but varies at every rate of flow—an undesirable characteristic.

The practical result of the foregoing equations is that a flow controller receiving an impulse from the square-root transmitter can be stabilized equally well at all rates of flow. However, if the controller receiving an impulse from a differential transmitter were stabilized at high flows, it would be sluggish at low flows. If it were stabilized at low flows, it would be unstable at high flows. Of course, since unstability cannot be tolerated, the only choice is the former.

A particular case which illustrates the usefulness of this device is in a three-element feedwater control. Here the controller receives the combined impulse of feedwater flow, steam flow, and drum level. If the two flow functions were of the differential type, the relationship of the change in drum level to the change in flow would differ at every rate of flow. However, by use of the square-root-type transmitter this relationship is constant, and effective drum-level control can be obtained from full load on down to very low loads.

[3] Project Engineer, Republic Flow Meters Company, Chicago, Ill. Jun. ASME.

Author's Closure

The author appreciates Mr. Perry's mention of the simplified stabilizing procedure in the field when the instrument described above is used.

The amount of effort put into the development of square-root extracting devices by designers through the years confirms Mr. Perry's appraisal of their value.

On the Solution of the Reynolds Equation for Slider-Bearing Lubrication—III

Effect of Transverse Curvature[1]

By A. CHARNES,[2] E. SAIBEL,[3] and A. S. C. YING,[4] PITTSBURGH, PA.

The effect of transverse curvature on the properties of a rectangular slider bearing with side leakage is studied for film thickness varying exponentially in the direction of motion and symmetrically perpendicular to this direction. A perturbation technique allows the expression of the results in terms of the same type of simple solution previously obtained without transverse curvature. Thus the solution is convenient for all length to width ratios and entrance to exit clearances.

Introduction

THE problem of slider-bearing lubrication with side leakage for film thickness varying exponentially in the direction of motion has been considered in a previous paper.[5] However, the film thickness has been taken as constant in the transverse direction (perpendicular to motion). Little information seems to be available on the slider-bearing problem with film thickness varying in both directions, although this is of great importance especially for small curved film thickness in the transverse direction.

In this paper the slight variation of film thickness in the transverse direction is taken care of by a perturbation of the solution obtained in the previous paper. Due to the simplicity of the previous solution, the present result turns out to be quite tractable and is valid for any ratio of length to width of slider, or for any ratio of entrance to exit clearance.

The notation used is indicated in Fig. 1. As usual, p denotes the pressure and h the film thickness. The assumptions made are those which lead to the Reynolds equation with constant viscosity.

Theory of Perturbation

The Reynolds equation for fluid lubrication is

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(h^3 \frac{\partial p}{\partial y}\right) = 6\mu U \frac{\partial h}{\partial x} \quad [1]$$

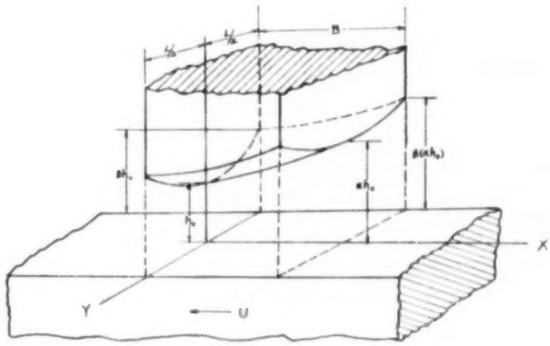


Fig. 1

Let

$$h^3 \equiv H \quad [2]$$

and

$$6\mu U \frac{\partial h}{\partial x} \equiv g(x, y) \quad [3]$$

Then Equation [1] may be written

$$\frac{\partial}{\partial x}\left(H \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(H \frac{\partial p}{\partial y}\right) = g(x, y) \quad [4]$$

For $h = h_0(x)$, suppose the solution has been previously found. Designate it by $p = p_0(x, y)$.

Now assume

$$\left.\begin{aligned} H &= H_0 + \lambda H_1 + \lambda^2 H_2 + \dots \\ g &= g_0 + \lambda g_1 + \lambda^2 g_2 + \dots \\ p &= p_0 + \lambda p_1 + \lambda^2 p_2 + \dots \end{aligned}\right\} \quad [5]$$

Substituting these into Equations [4], we have

$$\frac{\partial}{\partial x}\left[(H_0 + \lambda H_1 + \lambda^2 H_2 \dots)\left(\frac{\partial p_0}{\partial x} + \lambda \frac{\partial p_1}{\partial x} + \lambda^2 \frac{\partial p_2}{\partial x} + \dots\right)\right]$$
$$+ \frac{\partial}{\partial y}\left[(H_0 + \lambda H_1 + \lambda^2 H_2 + \dots)\left(\frac{\partial p_0}{\partial y} + \lambda \frac{\partial p_1}{\partial y} + \lambda^2 \frac{\partial p_2}{\partial y} + \dots\right)\right] = g_0 + \lambda g_1 + \lambda^2 g_2 + \dots$$

Equating coefficients of like powers of λ, we find for the terms of zero order

$$\frac{\partial}{\partial x}\left(H_0 \frac{\partial p_0}{\partial x}\right) + \frac{\partial}{\partial y}\left(H_0 \frac{\partial p_0}{\partial y}\right) = g_0 \quad [6]$$

The solution of this equation, as already indicated, is known and designated by

[1] The research underlying this paper was partially supported by funds from a U. S. Air Force contract with the Carnegie Institute of Technology.

[2] Associate Professor, Department of Mathematics, Carnegie Institute of Technology.

[3] Professor of Mechanics, Department of Mathematics, Carnegie Institute of Technology. Mem. ASME.

[4] Teaching Assistant, Department of Mechanical Engineering, Carnegie Institute of Technology. Jun. ASME.

[5] "On the Solution of the Reynolds Equation for Slider-Bearing Lubrication—I," by A. Charnes and E. Saibel, Trans. ASME, vol. 74, 1952, pp. 867–873.

Contributed by the Lubrication Research Committee and Machine Design Division and presented at the Fall Meeting, Chicago, Ill., September 8–11, 1952, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, June 10, 1952. Paper No. 52—F-37.

$$p_0 = p_0(x, y)$$

The terms of the first order in λ yield

$$\frac{\partial}{\partial x}\left(H_1\frac{\partial p_0}{\partial x} + H_0\frac{\partial p_1}{\partial x}\right) + \frac{\partial}{\partial y}\left(H_1\frac{\partial p_0}{\partial y} + H_0\frac{\partial p_1}{\partial y}\right) = g_1$$

or

$$\frac{\partial}{\partial x}\left(H_0\frac{\partial p_1}{\partial x}\right) + \frac{\partial}{\partial y}\left(H_0\frac{\partial p_1}{\partial y}\right) = g_1 - \left[\frac{\partial}{\partial x}\left(H_1\frac{\partial p_0}{\partial x}\right) + \frac{\partial}{\partial y}\left(H_1\frac{\partial p_0}{\partial y}\right)\right] \quad [7]$$

We can solve for p_1 by knowing p_0.

Similarly

$$\frac{\partial}{\partial x}\left(H_2\frac{\partial p_0}{\partial x} + H_1\frac{\partial p_1}{\partial x} + H_0\frac{\partial p_2}{\partial x}\right) + \frac{\partial}{\partial y}\left(H_2\frac{\partial p_0}{\partial y} + H_1\frac{\partial p_1}{\partial y} + H_0\frac{\partial p_2}{\partial y}\right) = g_2 \quad [8]$$

We can solve for p_2 by knowing p_0 and p_1, etc.

The final solution of Equation [4] is

$$p = p_0 + \lambda p_1 + \lambda^2 p_2 + \ldots$$

If λ is small

$$p = p_0 + \lambda p_1$$

Assume

$$h = ae^{bx}e^{cy^2} \quad [9]$$

where a, b, and c are known constants determined from the shape of slider. Then

$$H = h^3 = a^3e^{3bx}e^{3cy^2}$$

$$= a^3e^{3bx}\left[1 + 3cy^2 + \frac{(3cy^2)^2}{2!} + \ldots\right] = H_0 + \lambda H_1 + \ldots$$

$$g = 6\mu U\frac{\partial h}{\partial x} = 6\mu Uabe^{bx}e^{cy^2}$$

$$= 6\mu Uabe^{bx}\left[1 + cy^2 + \frac{(cy^2)^2}{2!} + \ldots\right] = g_0 + \lambda g_1 + \ldots$$

where

$$\lambda = 3c,\ H_0 = a^3e^{3bx},\ H_1 = y^2H_0$$

$$g_0 = 6\mu Uabe^{bx},\ g_1 = \frac{y^2}{3}g_0$$

From Equation [6] we obtain

$$p_0 = \sum_{n=1}^{\infty} A_n e^{-\frac{3b}{2}x}\sin\frac{n\pi}{B}x\cosh\sqrt{\frac{n^2\pi^2}{B^2} + \frac{9b^2}{4}}\,y + f(x)$$

Substituting these into Equation [7] we have

$$\frac{\partial^2 p_1}{\partial x^2} + \frac{\partial^2 p_1}{\partial y^2} + 3b\frac{\partial p_1}{\partial x} = -\frac{4\mu Ub}{a^2}y^2e^{-2bx} - 2ye^{-\frac{3b}{2}x}\sum_{n=1}^{\infty} A_nk_n\sin\frac{n\pi x}{B}\sinh k_ny \quad [10]$$

where

$$k_n = \sqrt{\frac{n^2\pi^2}{B^2} + \frac{9b^2}{4}} \quad [11]$$

with the boundary conditions $p = 0$ on the edges of the slider.

Solution for Perturbation Term

The solution of Equation [10] consists of two parts, namely, the particular integral p_p, and the complementary function p_c. It will be shown that the particular integral is so chosen that the boundary conditions for the complementary function will be as simple as possible. Actually, what we do is make $p_p = 0$ at $y = \pm L/2$, and then $p_c = 0$ at $y = \pm L/2$ and $p_c = -p_p$ at $x = 0$ and B.

(*a*) *The Particular Integral.* For the first term on the right-hand side of Equation [10]

$$\frac{\partial^2 p_1}{\partial x^2} + \frac{\partial^2 p_1}{\partial y^2} + 3b\frac{\partial p_1}{\partial x} = -\frac{4\mu Ub}{a^2}y^2e^{-2bx} \quad [12]$$

The solution will be in the form

$$p_1 = e^{-2bx}F(y) \quad [13]$$

Substituting this into Equation [12], we have the differential equation for $F(y)$

$$F''(y) - 2b^2F(y) = -\frac{4\mu Ub}{a^2}y^2$$

and its solution is

$$F(y) = C_1e^{\sqrt{2}by} + C_2e^{-\sqrt{2}by} + \frac{2\mu U}{a^2b}y^2 + \frac{2\mu U}{a^2b^3} \quad [14]$$

where the first two terms are the complementary function and the last two terms are the particular integral of $F(y)$.

Using the boundary conditions $p_1 = 0$ or $F(y) = 0$ at $y = \pm L/2$, we find

$$C_1 = C_2 = -\frac{2\mu U}{a^2b^3}\,\frac{1 + \left(\frac{bL}{2}\right)^2}{e^{\sqrt{2}b\frac{L}{2}} + e^{-\sqrt{2}b\frac{L}{2}}} = \frac{C}{2} \quad [15]$$

For the series part of the right-hand side of Equation [10], we shall find the particular solution for one typical term only, from

$$\frac{\partial^2 p_1}{\partial x^2} + \frac{\partial^2 p_1}{\partial y^2} + 3b\frac{\partial p_1}{\partial x} = -2A_nk_nye^{-\frac{3b}{2}x}\sin\frac{n\pi}{B}x\sinh k_ny$$

$$= -\frac{A_nk_n}{2i}y\left[e^{\left(\frac{-3b}{2}+\frac{n\pi i}{B}\right)x}e^{k_ny} - e^{\left(\frac{-3b}{2}+\frac{n\pi i}{B}\right)x}e^{-k_ny} - e^{\left(\frac{-3b}{2}-\frac{n\pi i}{B}\right)x}e^{k_ny} + e^{\left(\frac{-3b}{2}-\frac{n\pi i}{B}\right)x}e^{-k_ny}\right] \quad [16]$$

For the first term the solution will be in the form

$$p_1 = e^{\left(\frac{-3b}{2}+\frac{n\pi i}{B}\right)x}e^{k_ny}\phi(y)$$

Substituting this into Equation [16], we have the equation for $\phi(y)$

$$\phi''(y) + 2k_n\phi'(y) = -\frac{A_nk_n}{2i}y$$

and its solution is

$$\phi(y) = -\frac{A_nk_n}{2i}\left(\frac{1}{4k}y^2 - \frac{1}{4k_n^2}y + C\right)$$

Similarly, for other three terms in Equation [16], the corresponding parts of the particular solution are

$$e^{\left(\frac{-3b}{2}-\frac{n\pi}{B}i\right)x}\, e^{k_n y}\, \phi(y),\; e^{\left(\frac{-3b}{2}+\frac{n\pi i}{B}\right)x}\, e^{-k_n y}\, \psi(y)$$

and

$$e^{\left(\frac{-3b}{2}-\frac{n\pi i}{B}\right)x}\, e^{-k_n y}\, \psi(y)$$

where

$$\psi(y) = -\frac{A_n k_n}{2i}\left(-\frac{1}{4k_n}y^2 - \frac{1}{4k_n^2}y + C'\right)$$

By summing up and rearranging we have the complete particular solution for the series

$$p_1 = -\frac{A_n k_n}{2} e^{\frac{-3b}{2}x} \sin\frac{n\pi}{B} x\, G_n(y) \qquad [17]$$

where

$$G_n(y) = \frac{1}{k_n^2} y^2 \cosh k_n y - \frac{1}{k_n^2} y \sinh k_n y$$
$$+ K' \sinh k_n y + K \cosh k_n y \qquad [18]$$

As before, using the boundary conditions $p_1 = 0$ or $G_n(y) = 0$ at $y = \pm L/2$, results in $K' = 0$ and

$$K = \frac{L}{4k_n^2} \tanh \frac{k_n L}{2} - \frac{L^2}{8} \qquad [19]$$

The complete particular solution for Equation [10] is

$$p_p = e^{-2bx}F(y) - \sum_{n=1}^{\infty} A_n e^{-\frac{3b}{2}x} \sin\frac{n\pi}{B} x\, G_n(y) \qquad [20]$$

where $F(y)$ and $G_n(y)$ refer to Equations [14] and [18].

(*b*) Complementary Function

$$\frac{\partial^2 p_1}{\partial x^2} + \frac{\partial^2 p_1}{\partial y^2} + 3b\frac{\partial p_1}{\partial x} = 0 \qquad [21]$$

A solution by separation of variables will be obtained in the usual way by assuming

$$p_c = X(x)\, Y(y)$$

This leads to

$$X'' + 3bX' - k^2 X = 0 \qquad [22]$$

and

$$Y'' + k^2 Y = 0 \qquad [23]$$

where k is a constant to be determined. The solutions of these equations are, respectively

$$X = e^{\frac{-3bx}{2}} (E' \sinh \xi x + F' \cosh \xi x)$$

where

$$\xi = \sqrt{\frac{9b^2}{4} + k^2}$$

and

$$Y = G \sin ky + H \cos ky$$

Because of symmetry in y, $G = 0$; also, $p = 0$ when $y = L/2$. This is possible for a nonzero solution if

$$\frac{kL}{2} = \frac{\pi}{2}(2m - 1) \qquad [24]$$

or

$$k_m = \frac{\pi}{L}(2m-1) \quad \text{and} \quad \xi = \sqrt{\frac{9b^2}{4} + \frac{\pi^2}{L^2}(2m-1)^2}$$

m being an integer.

The complementary function takes the form

$$p_c = \sum_{m=1}^{\infty} e^{\frac{-3b}{2}x} \cos(2m-1)\frac{\pi y}{L}(E_m \sinh \xi x + F_m \cosh \xi x) \qquad [25]$$

The E_m and F_m will now be determined from the boundary conditions that on $x = 0$, $0 = p = p_p + P_c$. This means $p_c = -p_p = -F(y)$ and that on $x = B$; $0 = p = p_p + P_c$ means $p_c = -p_p = -e^{-2bB}F(y)$. Thus we find

$$F_m = -\frac{4}{L}\int_0^{L/2} F(y) \cos(2m-1)\frac{\pi y}{L}\, dy$$

$$= (-1)^m \frac{4}{L}\left\{ C \frac{(2m-1)\frac{\pi}{L}\cosh\sqrt{2}\, b\frac{L}{2}}{2b^2 + \frac{(2m-1)^2\pi^2}{L^2}} \right.$$

$$\left. + \frac{2\mu U}{a^2 b}\left[\frac{\frac{L^2}{4}}{(2m-1)\frac{\pi}{L}} - \frac{2}{(2m-1)^3\frac{\pi^3}{L^3}} + \frac{1}{b^2(2m-1)\frac{\pi}{L}}\right]\right\}$$

and

$$E_m = \frac{e^{\frac{-bB}{2}} - \cosh \xi B}{\sinh \xi B} F_m \qquad [26]$$

The final solution for p_1 is

$$p_1 = e^{-2bx} F(y) - \sum_{n=1}^{\infty} A_n e^{\frac{-3b}{2}x} \sin\frac{n\pi}{B} G_n(y)$$

$$+ \sum_{m=1}^{\infty} e^{\frac{-3b}{2}x} \cos(2m-1)\frac{\pi y}{L}(E_m \sinh \xi x + F_m \cosh \xi x) \qquad [27]$$

where $F(y)$, $G_n(y)$, E_m, and F_m are given in Equations [14], [18], and [26]. A_n is the same expression as was used in the previous paper.[3]

Performance Characteristics

The characteristics treated in the present paper are (*a*) total load and (*b*) the frictional force. Other characteristics such as center of pressure, oil flow, and so on, can be obtained from the solution given by Equation [27].

(*a*) *Total Load.* Since the slope of the slider is small, it is reasonable to assume that the vertical component of the pressure against the slider is equal to the pressure itself; thus we have

$$W = \int_{y=-L/2}^{L/2} \int_{x=0}^{B} p \, dx \, dy \qquad [28]$$

$$= \int_{y=-L/2}^{L/2} \int_{x=0}^{B} (p_0 + \lambda p_1) \, dx \, dy$$

$$= \int_{y=-L/2}^{L/2} \int_{x=0}^{B} p_0 \, dx \, dy + \lambda \int_{y=-L/2}^{L/2} \int_{x=0}^{B} p_1 \, dx \, dy$$

$$= W_0 + \lambda W_1$$

where

$$W_0 = \int_{y=-L/2}^{L/2} \int_{x=0}^{B} p_0 \, dx \, dy$$

(as calculated in the previous paper)[5]
and

$$W_1 = \int_{y=-L/2}^{L/2} \int_{x=0}^{B} p_1 \, dx \, dy \qquad [29]$$

where p_1 is given by Equation [27]. This results in

$$W_1 = \frac{e^{-3bB} - 1}{-b} \left(\frac{C}{\sqrt{2}\, b} \sinh \sqrt{2}\, b \frac{L}{2} + \frac{\mu U L^3}{12 a^2 b} + \frac{\mu U L}{a^2 b^3} \right)$$

$$-2 \sum_{n=1}^{\infty} A_n \frac{-\frac{n\pi}{B} \left[(-1)^n e^{-\frac{3bB}{2}} \right]}{\frac{9b^2}{4} + \frac{n^2\pi^2}{B^2}} \left[\left(\frac{L}{4k^2} \tanh \frac{kL}{2} \right.\right.$$

$$\left.\left. + \frac{3}{2k^3} \right) \sinh \frac{kL}{2} - \frac{3L}{4k^2} \cosh \frac{kL}{2} \right]$$

$$+ \sum_{m=1}^{\infty} F_m \left\{ \frac{e^{\frac{-bB}{2}} - \cosh \xi B}{\sinh \xi B} \left[\frac{e^{\left(-\frac{3b}{2}+\xi\right)B} - 1}{\frac{3b}{2} + \xi} \right.\right.$$

$$\left. - \frac{e^{\left(-\frac{3b}{2}-\xi\right)B} - 1}{-\frac{3b}{2} - \xi} \right] + \left[\frac{e^{\left(-\frac{3b}{2}+\xi\right)B} - 1}{-\frac{3b}{2} + \xi} \right.$$

$$\left.\left. + \frac{e^{\left(-\frac{3b}{2}-\xi\right)B} - 1}{-\frac{3b}{2} - \xi} \right] \right\} \frac{(-1)^{m+1}}{(2m-1)\frac{\pi}{L}} \qquad [30]$$

(b) *Frictional Force.* This can be calculated as follows

$$F = \int_{y=-L/2}^{L/2} \int_{x=0}^{B} s \, dx \, dy$$

where

$$s = \frac{\mu U}{h} + \frac{h}{2} \frac{\partial p}{\partial x}$$

Now assume

$$1/h = h' = h_0' + \lambda h_1' + \lambda^2 h_2' + \dots$$

$$s = \mu U (h_0' + \lambda h_1' + \lambda^2 h_2' + \dots) + \frac{1}{2} (h_0 + \lambda h_1$$

$$+ \lambda^2 h_2 + \dots) \left(\frac{\partial p_0}{\partial x} + \frac{\partial p_1}{\partial x} \dots \right) = \left(\mu U h_0' + \frac{h_0}{2} \frac{\partial p_0}{\partial x} \right)$$

$$+ \lambda \left(\mu U h_1' + \frac{h_1}{2} \frac{\partial p_0}{\partial x} + \frac{h_0}{2} \frac{\partial p_1}{\partial x} \right) + \lambda^2 (\quad) + \dots$$

$$= s_0 + \lambda s_1 + \lambda^2 s_2 + \dots$$

For small λ

$$s = s_0 + \lambda s_1$$

where

$$s_0 = \mu U h_0' + \frac{h_0}{2} \frac{\partial p_0}{\partial x}$$

and

$$s_1 = \mu U h_1' + \frac{h_1}{2} \frac{\partial p_0}{\partial x} + \frac{h_0}{2} \frac{\partial p_1}{\partial x}$$

$$F \doteq \int_{y=-L/2}^{L/2} \int_{x=0}^{B} s_0 \, dx \, dy + \lambda \int_{y=-L/2}^{L/2} \int_{x=0}^{B} s_1 \, dx \, dy$$

which may be written

$$F = F_0 + \lambda F_1 \qquad [31]$$

It is evident that F_0 is the frictional force obtained previously[5] and F_1 is

$$F_1 = \int_{y=-L/2}^{L/2} \int_{x=0}^{B} \left(\mu U h_1' + \frac{h_1}{2} \frac{\partial p_0}{\partial x} + \frac{h_0}{2} \frac{\partial p_1}{\partial x} \right) dx \, dy \qquad [32]$$

As before, assume

$$h = a e^{bx} e^{-cy^2}$$

$$h_0 = a e^{bx}$$

$$h_1 = \frac{y^2}{3} a e^{bx}$$

and

$$h' = \frac{1}{h} = \frac{1}{a} e^{-bx} e^{-cy^2} = \frac{1}{a} e^{-bx} \left(1 + \frac{-cy^2}{1!} + \dots \right)$$

$$= h_0' + \lambda h_1' + \dots$$

where

$$h_0' = \frac{1}{a} e^{-bx}, \quad \lambda = 3c$$

and

$$h_1 = -\frac{y^2}{3} \frac{1}{a} e^{-bx}$$

Substituting the foregoing and using Equation [27] for p_1, we have

$$F_1 = \int_{y=-L/2}^{L/2} \int_{x=0}^{B} \left[\mu U \left(-\frac{y^2}{3} \frac{1}{a} e^{-bx} \right) + \frac{y^2}{3} a e^{bx} \frac{\partial p_0}{\partial x} \right.$$

$$\left. + a e^{bx} \frac{\partial p_1}{\partial x} \right] dx \, dy = \frac{\mu U}{36 b a} L^3 (e^{-bB} - 1)$$

$$- \frac{ab}{3} \sum_{n=1}^{\infty} A_n \frac{-\frac{n\pi}{B}}{\frac{b^2}{4} + \frac{n^2\pi^2}{B^2}} \left[(-1)^n e^{-\frac{bB}{2}} - 1 \right] \times$$

$$\left[\left(\frac{L^2}{4k_n} + \frac{2}{k_n^3} \right) \sinh \frac{k_n L}{2} - \frac{L}{k_n^2} \cosh \frac{k_n L}{2} \right] - \dots \qquad [33]$$

$$\frac{abL^2}{72}\left[\frac{3\mu U}{a^3b^2}(e^{-bB}-1)-\frac{k_1}{2b}(e^{-2bB}-1)+\frac{k_2}{b}(e^{bB}-1)\right]-$$

$$ab\,\frac{e^{-bB}-1}{-b}\int_0^{L/2}F(y)dy+$$

$$\frac{1}{ab}\sum_{n=1}^{\infty}A_n\frac{-\frac{n\pi}{B}\left[(-1)^n e^{-\frac{bB}{2}}-1\right]}{\frac{b^2}{4}+\frac{n^2\pi^2}{B^2}}\int_0^{L/2}G_n(y)dy-\frac{ab}{2}\sum_{m=1}^{\infty}F_m\left\{\frac{e^{-\frac{bB}{2}}-\cosh\xi B}{\sinh\xi B}\left[\frac{e^{\left(-\frac{b}{2}+\xi\right)B}-1}{-\frac{b}{2}+\xi}\right.\right.$$

$$\left.\left.-\frac{e^{\left(-\frac{b}{2}-\xi\right)B}-1}{-\frac{b}{2}-\xi}\right]+\left[\frac{e^{\left(-\frac{b}{2}+\xi\right)B}-1}{-\frac{b}{2}+\xi}+\frac{e^{\left(-\frac{b}{2}-\xi\right)B}-1}{-\frac{b}{2}-\xi}\right]\right\}\frac{(-1)^{m+1}}{(2m-1)\frac{\pi}{L}} \qquad [33]\ (cont.)$$

where

$$\left.\begin{aligned}\int_0^{L/2}F(y)dy &= \frac{C}{\sqrt{2}\,b}\sinh\sqrt{2}\,b\,\frac{L}{2}+\frac{\mu UL}{a^3b^3}+\frac{\mu UL^3}{12a^3b}\\ &\text{and}\\ \int_0^{L/2}G_n(y)dy &= \left(\frac{L}{4k_n^2}\tanh k_n\frac{L}{2}+\frac{3}{2k_n^3}\right)\sinh\frac{k_nL}{2}-\frac{3L}{4k_n^2}\cosh\frac{k_nL}{2}\end{aligned}\right\}\ldots[34]$$

Numerical Examples

As an example of the use of the method, we calculate the total load and frictional force for a square slider having a ratio of film thickness at entrance to exit of 2 and a ratio of film thickness at side to center of 1.02, Fig. 1.

Let

$$\frac{h_{x=B}}{h_{x=0}}=\alpha \quad\text{and}\quad \frac{h_{y=\pm\frac{L}{2}}}{h_{y=0}}=\beta$$

$$h=h_0 \text{ at } x=0,\ y=0$$

and

$$h=\alpha h_0 \text{ at } x=B,\ y=0$$

$$h=\beta h_0 \text{ at } x=0,\ y=\pm\frac{L}{2}$$

$$h=\alpha(\beta h_0) \text{ at } x=B,\ y=\pm\frac{L}{2}$$

Then from $h = ae^{bx}e^{cy^2}$, we have

$$a=h_0$$

$$b=\frac{\log_e\alpha}{B}$$

$$c=\frac{4\log_e\beta}{L^2}$$

for $\alpha = 2$, $\beta = 1.02$

$$b=\frac{0.693}{B},\quad\text{and}\quad c=\frac{0.0730}{L^2}\left(\lambda=\frac{0.219}{L^2}\right)$$

and $L = B$ for square slider.

From these values we obtain

$$W_1 = 0.000985\,\frac{6\mu UB^3}{h_0^3}$$

From previous work[5]

$$W_0 = -0.01130\,\frac{6\mu UB^3}{h_0^3}$$

for the same case. Thus

$$W = W_0+\lambda W_1 = (-0.01130+0.219\times 0.000985)\,\frac{6\mu UB^3}{h_0^3}$$

$$= -0.01109\,\frac{6\mu UB^3}{h_0^3}$$

Substituting these values into Equation [33], we get

$$F_1 = -0.00371\,\frac{6\mu UB^2}{h_0}$$

and from the previous paper[5]

$$F_0 = 0.1259\,\frac{6\mu UB^2}{h_0}$$

for the same case. Then

$$F = F_0+\lambda F_1 = (0.1259-0.219\times 0.00371)\,\frac{6\mu UB^2}{h_0}$$

$$= 0.1250\,\frac{6\mu UB^2}{h_0}$$

The result indicates that both the total load and frictional force decrease for a convex slider.

Conclusion

It is reasonable to assume that the perturbation method yields quite accurate results because both for total load and the frictional force the perturbation term is small in comparison with the first term. It is less than 2 per cent for the former and less than 1 per cent for the latter when $\beta = 1.02$. Now, it is interesting to investigate the range in which this method can be applied. For this reason, two curves, $\lambda W_1/W_0$ (ratio of perturbation term to original value) against β, and $\lambda F_1/F_0$ against β are plotted, Figs. 2 and 3.

It should be noted that in plotting the two curves W_1 and F_1 remain constant; only the values of λ need be calculated. This can easily be carried out and the results are listed in Tables 1 and 2.

The method appears to be quite useful for side to center clearance up to 1.40, or a ratio of the perturbed term to original one of about 10 per cent.

The foregoing examples were calculated for the convex slider; that is, the side clearance is larger than that at the center. Actu-

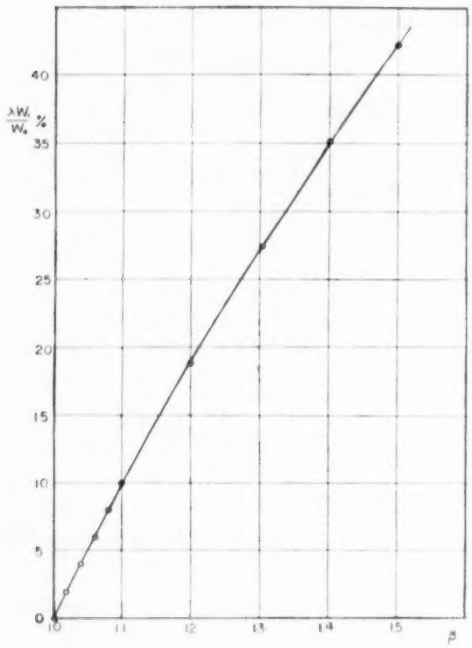


Fig. 2

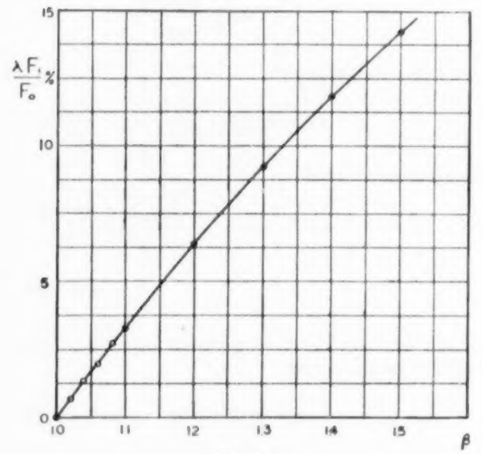


Fig. 3

ally, the method derived can be applied to the concave slider as well, the only difference being in the sign of λ. It is negative in the concave case.

REFERENCE

"Lubrication," by A. E. Norton, McGraw-Hill Book Company, Inc., New York, New York, 1942.

Discussion

F. Osterle.[6] In this paper the unperturbed solution is based on

TABLE 1

β	λL^2	$\frac{\lambda W_1}{W_0}$, per cent	$W = W_0 + \lambda W_1$ in $\frac{6\mu UB^3}{h_0^2}$
1	0	0	0.01130
1.02	0.219	1.91	0.01109
1.04	0.470	4.10	0.01082
1.06	0.700	6.10	0.01060
1.08	0.924	8.05	0.01037
1.10	1.142	9.96	0.00965
1.20	2.19	19.10	0.00914
1.30	3.15	27.4	0.00820
1.40	4.03	35.2	0.00732
1.50	4.86	42.3	0.00652

TABLE 2

β	λL^2	$\frac{\lambda F_1}{F_0}$, per cent	$F = F_0 + \lambda F_1$ in $\frac{6\mu UB^2}{h_0}$
1.0	0	0	0.1259
1.02	0.219	0.644	0.1250
1.04	0.470	1.388	0.1240
1.06	0.700	2.07	0.1231
1.08	0.924	2.73	0.1225
1.10	1.142	3.36	0.1216
1.20	2.19	6.44	0.1180
1.30	3.15	9.26	0.1141
1.40	4.03	11.90	0.1110
1.50	4.86	14.35	0.1074

the assumption that the slider is not plane but is curved exponentially in the direction of motion. This curvature is small so that, as was shown in the authors' earlier paper,[7] in most cases little disagreement exists between results obtained for plane sliders by the Michell solution and those obtained for exponential sliders by the authors' solution. However, the authors' use of the perturbation technique to put in slider curvature in the direction transverse to motion suggests the possibility that, if it ever becomes necessary to do so, the same technique could be used to take out slider curvature in the direction of motion. The exponential film thickness given by

$$h = ae^{bx} = a\left(1 + bx + \frac{b^2x^2}{2} + \ldots\right)$$

is, in most cases, adequately represented by the first three terms of the series. To perturb the film thickness into the linear case it would be necessary to subtract the term $b^2x^2/2$.

In the authors' investigation of the range of transverse curvatures over which the perturbation technique can be applied the criterion is employed that as long as the amount by which the load or friction force is corrected is less than 10 per cent of the original value of the load or friction force the method yields acceptable results. The writer wonders why 10 per cent is used as the criterion. A possible reason is that if the first correction term is 10 per cent of the original and the assumption is made that the second correction term will be on the order of 10 per cent of the first correction term and so on, an accuracy of roughly 1 per cent could be claimed.

A. A. Raimondi.[8] The authors once again have demonstrated the versatility of depicting film shapes of rectangular slider bearings by exponential formulas. This powerful method can no doubt be applied to many more problems and it is hoped that the authors will be encouraged to continue along these lines.

In attempting to interpret the data given with consideration to practical significance, it can be assumed that increasing values of the parameter β would mean increasing values of the amount of deflection of an originally flat slider, or pad. This deflection is interpreted to occur (in the practical case of a pivoted-pad thrust bearing) in the radial direction. The authors have shown that for a radial deflection of the order of 150 per cent of the minimum film thickness ($\beta = 1.50$) a decrease in load-carrying capacity of about 42 per cent should be expected.

[6] Mechanical Engineering Department, Carnegie Institute of Technology Pittsburgh, Pa. Jun. ASME.

[7] Authors' reference 5.

[8] Research Engineer, Westinghouse Electric Company, East Pittsburgh, Pa. Jun. ASME.

It would have been interesting if the authors had also shown the effect of increasing values of β on the location of the center of pressure, that is, the pivot position. It is the writer's estimate that due to increased side leakage (and the accompanying alteration in the pressure distribution) with increasing β, the center of pressure would retreat toward the trailing edge of the slider. This leads to the interesting fact that sliders deformed in this manner would not tend to operate as well as flat sliders when the pivot is placed midway between the leading and trailing edge, that is, centrally located. This conclusion is reached on the premise that the main reason a slider bearing operates satisfactorily with its pivot centrally located is due to the decrease in viscosity of the lubricating liquid as it passes across the slider. This seems to be the most popular explanation at the present time although it has been observed that Fogg's thermal-wedge effect may contribute to the part played by variable viscosity when a liquid lubricant is used. Boswall[9] has shown that rounding the pad entrance has the same effect.

It also has been stated in the literature that an air-lubricated bearing with the pivot centrally located will operate satisfactorily. Since the viscosity of air increases with temperature the variable-viscosity theory will not explain the satisfactory performance. Shaw[10] shows that Fogg's thermal-wedge effect will not explain it either. Hence there is some unknown factor which plays an important role in explaining the satisfactory operation of centrally pivoted tilting-pad thrust bearings.

A factor which may well be important and which has not received much consideration as yet is the shape or configuration of the pad itself. In practical applications, the pads also may deform about the pivot point in the longitudinal direction, that is, in the direction of motion, as well as in the radial direction which this paper implies. If this deformation is such that a diverging oil film exists over some region of the pad near the trailing edge, the oil film will rupture in this region and the center of pressure will move toward the entrance end. The amount of advance of the center of pressure depends upon the nature and extent of this diverging region. In fact, calculations made by the writer's company with an electrical resistance network have shown that it is possible for the center of pressure to lie well in advance of the central position when the deformed slider has a spherical, or convex, configuration.

Central pivot-position operation, chamfering of entrance and exit sections, and pad shape are subjects which have intrigued the writer's associates. Much time and effort have been and are being devoted to their study. To further this study special equipment already has been built for producing accurately desired degrees of pad flatness, convexity, or concavity and for performance-testing these pads at different pivot positions. Some test data have been obtained and considerable time has been devoted to a mathematical evaluation. It is intended to publish these findings in the future.

[9] "The Theory of Film Lubrication," by R. O. Boswall, Longmans, Green & Company, Inc., New York, N. Y., 1928.

[10] "An Analysis of the Parallel-Surface Thrust Bearing," by M. C. Shaw, Trans. ASME, vol. 69, 1947, p. 387.

Authors' Closure

The suggestion of Dr. Osterle that the perturbation technique be used to obtain a better approximation to the plane slider from the solution for the exponential slider is an interesting one. This could certainly be done if closer agreement were desired between the two solutions. However, in view of the accuracy of the approximation as it now stands, it does not seem worth the effort.

In the present paper a value of 10 per cent was chosen arbitrarily since in application of perturbation methods to corresponding problems in other fields this is roughly the limit for good accuracy. It is usually quite difficult to establish exact bounds to the error.

Mr. Raimondi raises some very important and basic questions. The authors hope to show soon the effect of several of the variables on the location of the center of pressure as well as the effect of β on it. The latter can be calculated from the development in the present paper. It will be of great interest to the authors to study the results obtained by Mr. Raimondi and his associates in evaluating the relative importance of the various factors. The authors hope that this work will appear soon.

A Simple High-Speed Air Spinner for Centrifugal Testing of Small Mechanical Devices

By C. F. Bild[1] and P. F. Vial[2]

This centrifuge was developed for the purpose of testing the functioning of mechanical and electromechanical devices while these devices were spinning up to 1000 rps. A typical device is a clock mechanism, in which parts of considerable mass shift position during test.

Introduction

This project was started with the purpose of developing a simple laboratory spinner for checking time intervals of small timing devices at spins up to 36,000 rpm and for checking strength and operation of small devices at spins up to 120,000 rpm. It was also desired that this new spinner eliminate or minimize the bearing wear, balancing problems, and heating problems inherent in other types of spinners. Conventional-type spinners which utilize motor-armature shafts on rigidly mounted belt-driven spindles have proved very satisfactory at low spins, but require precise balancing for high-spin use. With rigid mounts at high speed, the shaft, test chuck, and parts being tested must be accurately balanced statically and dynamically to avoid excessive vibration, bearing wear, and heating. Even with accurate balancing, the bearings must be replaced frequently when high angular velocities are used. When testing developmental devices, whose design is being changed frequently, precise balancing becomes a time-consuming procedure.

Attention was directed toward adapting and modifying the air-driven spinning top[3] originated by French scientists and further developed and improved by Dr. J. W. Beams. It consists of a rotor supported and driven by air and is capable of high angular velocities, is relatively insensitive to static unbalance, and completely eliminates bearing problems. Thought also was given to the various types of vacuum-chamber spinners[4] utilizing air or magnetic drive and which were developed by Dr. Beams for spinning of large-diameter rotors at high speeds and under precise temperature control. It was decided, however, that these more complicated types were not needed and that the simpler spinning-top type, which does not require a vacuum chamber, would be satisfactory.

In the discussion of spinner design that follows, only those aspects which are new, or which have a direct bearing on the use of this spinner in testing small timing devices, will be covered. Details of construction, such as optimum rotor angle, number and size of flutes and jet size, have been discussed adequately in the reports by J. W. Beams,[3] W. D. Garman,[5] and K. R. May.[6]

First Successful Spinner

The initial problem was to develop an air spinner which could be used to test small timing devices (approximately 1.500 in. diam × 1.000 in. thick) at speeds up to 36,000 rpm. It was necessary that the device be taken up to speed in a locked condition, be unlocked after reaching speed, and that the timing interval from "instant of unlocking" to "run down" be measured. A means of braking the spinner in reasonable time and an accurate speed indicator were also needed.

It was decided that a rotor of approximately 3 in. diam would be needed. The diameter is an important factor in determining the maximum speed attainable and the angular acceleration for any given air pressure, both being higher for a small-diameter rotor. A large-diameter rotor is, however, more stable. It was soon found that with the air pressure available in our laboratory (100 psi) a $3^1/_4$-in-diam rotor easily reaches 36,000 rpm even when four spring-loaded $^1/_4$-in-diam carbon brushes are bearing on its top surface. Acceleration time to reach this speed is approximately 3 min.

The material of the rotor was also an important consideration. Acceleration time of a heavy rotor is considerably slower than that of a light rotor, but top speed attainable is little affected by weight. With a heavy rotor, more out-of-balance can be tolerated in the parts to be tested. For this latter reason, steel rotors have proved to be more suitable.

Proper cushioning of the stator in order to damp out vibrations is extremely important. With improper cushioning, the rotor will reach a critical speed, start chattering against the side of the stator, and stop accelerating. Setting the stator on a coil of soft flexible rubber hose has proved to be a satisfactory cushioning method.

When constructed properly, a spinning-top rotor is extremely stable. While the rotor is spinning, sizable weights can be shifted suddenly internally without visible effect on rotor stability. However, extreme static unbalance causes the rotor to chatter against the sides of the stator. Such a rotor also can be spun in an upside-down position, being held in the stator by the Bernoulli forces.

While large static unbalance is permissible, dynamic balance has to be maintained more closely. For this reason it is difficult to spin tall small-diameter rotors. Good rotor design requires that the moment of inertia, in the plane of rotation, be large. If this is done, precise balancing is not necessary. Properly designed rotors are extremely stable at high spins, and the driving air pressure can be changed suddenly without upsetting rotor stability.

Two methods have been used for making electrical connections between the parts inside the spinning rotor and the external

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[3] "The Production of High Rotational Speeds," by J. W. Beams and G. Pickels, *Review of Scientific Instruments*, vol. 6, 1935, pp. 299–308.

[4] "The Production and Maintenance of High Centrifugal Fields for Use in Biology and Medicine," by J. W. Beams, *Annals of New York Academy of Sciences*, October 6, 1942.

Contributed by the Machine Design Division and presented at the Fall Meeting, Chicago, Ill., September 8–11, 1952, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, July 3, 1952. Paper No. 52—F-34.

[5] "A Study of the High Speed Centrifuge," by W. D. Garman, *Review of Scientific Instruments*, vol. 4, 1933, pp. 450–453.

[6] "An Improved Spinning Top Homogeneous Spray Apparatus," by K. R. May, *Journal of Applied Physics*, vol. 20, 1949, pp. 932–938.

measuring instruments. One method was the use of concentric induction coils, two inside the rotor and two on stationary supports around the outside periphery of the rotor. Such a method proved successful but seemed to complicate the spinner construction unnecessarily; and, in addition, required auxiliary equipment such as oscillator and amplifier units and considerable shielding because of the weak signals involved.

The other method is the one now in use and consists of spring-loaded carbon brushes riding on phosphor-bronze contact rings on the top surface of the rotor. At first it was feared that the breaking effect resulting from brush pressure might prevent reaching the speeds desired, and it was found that brush pressure did increase acceleration time and decrease the top speed. However, by keeping the contact radius small and the brush pressure light, this did not become a serious problem. Brush speeds of approximately 10,000 fpm have been used successfully. Brush bounce, at high spins, is very noticeable on an oscilloscope but has not been sufficient to affect the operation of the timing circuit which uses an S-1 type[7] timing clock. If an electronic timer were used trouble might occur. In such a case, bounce could be reduced by increasing brush-spring pressure, by changing brush material, or by loading the brushes with a heavy powder to absorb energy.

Also, it was necessary in this application to have some method of initiating an action in the part being tested, after the rotor had reached operating speed. Again, two methods were tried. A magnetic release was dropped because of lack of space for the coil above the center of the rotor, this space being needed for the brush holders which had to be close to the center. In place of this, a small air piston was built into the top of the rotor so that a small jet of air shot at the center of the rotor caused the piston to move upward, initiating the action and closing an electrical contact at the same instant.

A brake was designed, which consists of a felt ring held in a bracket which can be pressed down to rub on the rotor top near the periphery. The rotor is stopped easily in this manner, and then the air is shut off. Breaking time from 36,000 rpm (600 rps) is approximately 30 sec.

It is desirable to incorporate an accurate method of speed indication on the spinner. This is accomplished by darkening one side of the rotor periphery chemically and polishing the other side. A beam of light is then focused on the rotor and reflected into a photoelectric cell. Each revolution the cell gives a pulse of voltage. These pulses are beat against a standard variable frequency in an oscilloscope to give the exact angular velocity of the rotor by means of Lissajou's figures. Speed is controlled by throttling the air pressure and can be held within 1 rps at the higher spins.

Figs. 1 to 5, inclusive, illustrate the type of spinner just described. A spinner of this type using a $3^1/_4$-in-diam rotor is being used daily in this laboratory for testing timing mechanisms at angular velocities up to 36,000 rpm (600 rps).

The spinner, as described, is an ideal instrument for laboratory developmental testing, but in its present form it is not suitable for rapid production testing.

The only maintenance that has been necessary on this spinner is the occasional removal of foreign particles that have lodged in the stator jets and partially blocked the flow of air. Such trouble is evidenced immediately by rotor chatter. This has occurred only twice in a period of 1 year and could be eliminated completely by filtering the air supply.

Miscellaneous Applications and Design Variations

Variations of this spinner have been put to many uses other than the timing of clock mechanisms. These include: stroboscopic inspection and study of detent and rotor actions, high-speed photographic studies of clock-escapement action under spin conditions, and off-center spinning of various mechanisms.

Much higher spins than 36,000 rpm (600 rps) can be attained by decreased rotor diameter, increased air pressure, or the elimination of brush-contact pressure. With slightly smaller-diameter rotors, much higher spins can be reached; for example, with 100 psi air pressure a 2.500-in-diam rotor will reach 66,000

[7] Standard Electric Time Company.

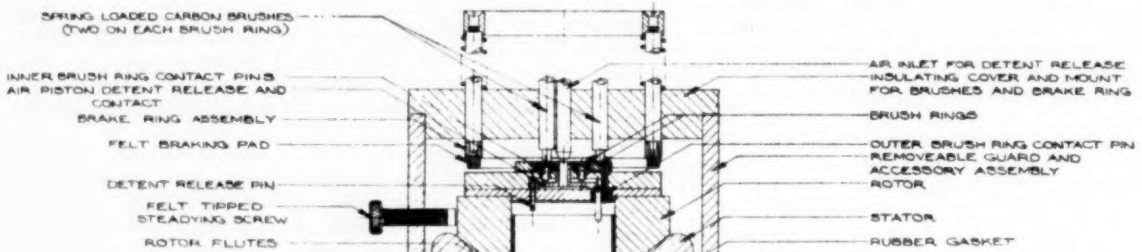

Fig. 1 Sectional View of High-Speed Air Spinner

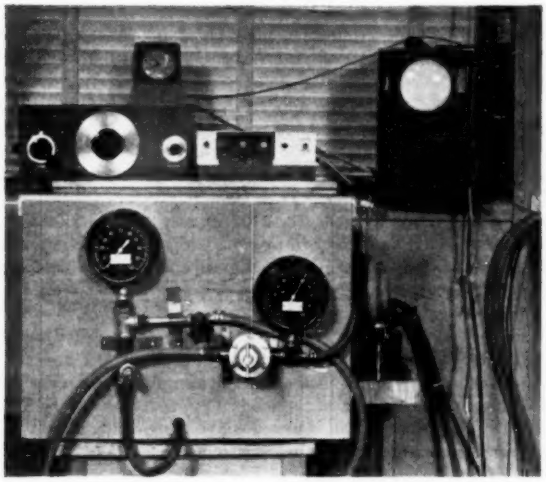

Fig. 2 High-Speed Air Spinner and Associated Equipment

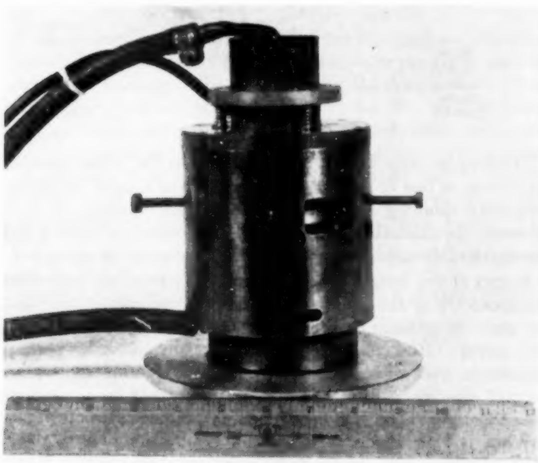

Fig. 3 Spinner Assembly

rpm and a 1.250-in-diam rotor will reach approximately 120,000 rpm. All these spins could be increased somewhat by increased air pressure. No attempt has been made to determine exact maximum speeds possible for various-diameter rotors; but Dr. Beams has done considerable work on determining the top spins of small-diameter rotors and references to his work are given.[3,4]

When long-duration testing at high spins is wanted, the air spinner is ideal, since there are no heating or bearing-wear problems. For example, components have been spun at 36,000 rpm (600 rps) continuously for 10 min. Parts could be spun continuously for hours just as easily.

Thought is being given to an inverted air spinner in which the stator is located above the rotor. When the air is turned on, the rotor would be sucked up into the stator and held there by the Bernoulli forces while it was spinning. This arrangement would simplify the construction and operation of the brake. Instead of a manually operated spring-loaded brake ring, a felt nest would be located just below the rotor, and the only action

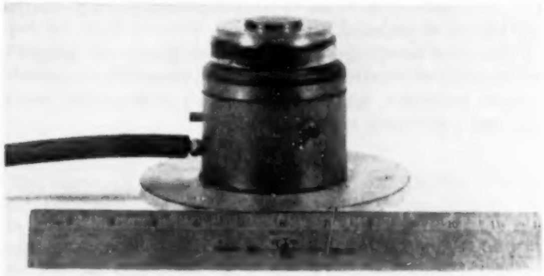

Fig. 4 Rotor in Place in Stator

Fig. 5 Stator, Rotor, and Rotor Cover

needed to stop the rotor would be to cut off the air supply. The rotor then would drop into the felt nest, and gravity would furnish the necessary pressure for rapid braking.

Conclusions

1 The spinning-top type of air spinner is ideally suited to high-spin developmental laboratory testing.

2 Clock-type mechanisms have been timed at spins of 36,000 rpm (600 rps) in rotors of $3^1/_4$ in. diam.

3 If considerably higher spins than 36,000 rpm are needed, they can be obtained by one or more of the following: Increasing air pressure, decreasing brush pressure, streamlining the rotor shape, polishing the rotor body, or decreasing rotor diameter. For example, a $2^1/_2$-in-diam rotor using 100 psi air pressure has been built which spins up to 66,000 rpm. Also, a rotor

of $1^1/_4$ in. diam can be made to reach approximately 120,000 rpm with 100 psi air pressure.

4 The chief advantages of this type of spinner are simplicity and economy of construction, versatility, relative insensitiveness to static unbalance, quietness, no heating problem, no bearing wear, and a minimum of maintenance.

Discussion

O. C. Brewster.[5] This paper describes an interesting application of the Beams spinning top. It is believed that many operations could take advantage of the possibility of very high rotative speed with the virtual absence of friction as exemplified by the Beams rotor or by using air (or other gas) as the lubricant in more or less conventional journal bearings. Kingsbury pointed this out before the turn of the century but his work has largely been forgotten and the writer knows of few practical applications of this type of lubrication. There is a fertile field of exploration here that is nearly untouched.

The writer recently has done a considerable amount of experimental work with air bearings. The minute one gets into the realm of really high speeds with air lubrication of plain cylindrical journal bearings, the phenomenon of "bearing whip" assumes great importance and under many conditions may block effectively the attainment of desired speeds.

Whip is particularly pronounced with gas lubrication because of the great fluidity of the lubricant and also because of its compressibility. It manifests itself as a vibration which increases in intensity rapidly as the speed increases and finally reaches such severity that higher speed becomes impossible.

A simple and effective method for eliminating bearing whip has been found.[6] By applying air pressure to one side of the bearing, thereby forcing the journal into an eccentric position in the bearing, whip is eliminated entirely. The amount of side air pressure required is a direct function of rotor speed and there appears to be no limit on the rotational speed obtainable other than that set by the limit of available air pressure, the strength of the rotor material, or the capabilities of the driving means.

As an illustration, a rotor was required for an optical device, the rotor to spin at 240,000 rpm (4000 rps). The exposed end of the rotor was to carry a mirror slanting slightly from the plane of rotation. Many attempts to use a Beams-type rotor were made but all were unsuccessful owing to instability. The problem was solved by the small air motor shown in Fig. 6 of this discussion. The steel rotor is $^1/_2$ in. diam $\times$ $2^3/_{16}$ in. long, running with 0.001-in. diametral clearance in two journal bearings $^1/_2$ in. long, spaced axially $^1/_2$ in. apart. An air turbine is provided, consisting of flutes milled into the rotor near one end, impelled by the jets from three small nozzles. Side air pressure is applied at the middle of each bearing through 0.025-in-diam holes.

With no side air pressure applied the rotor speed could not be forced higher than about 50,000 rpm, regardless of air supply to the turbine. At that speed the vibration was violent and would be destructive if allowed for any appreciable time; also, the speed fluctuated violently. With side air pressure applied this rotor attained speeds well over 240,000 rpm with no vibration. At this speed a minimum of 75-psig side air pressure was required to eliminate whip. A slight reduction in the pressure results in a sharply defined hum which increases with further reduction in pressure and is accompanied by a slowing down of the rotor owing to energy losses. Application of side air pressure higher than that required for the suppression of whip has no effect other than a very slight increase in friction due to reducing the thickness of the air film on the load-carrying side. A very high side air pressure would, of course, load the supporting film above its carrying capacity and the bearing would seize. The minimum side air-pressure requirements of this particular rotor are shown in Table 1 of this discussion.

TABLE 1 MINIMUM SIDE AIR PRESSURES

Rpm	Minimum side air press, psig	Turbine air press, psig
60000	15	7
120000	22	15
180000	30	25
240000	75	54

The largest rotor of this type with which the writer has any experience is one of 1 in. diam in the form of a small laboratory centrifuge spinning at 75,000 rpm but there appears to be no inherent size limitation, and there is every reason to believe that this method should be effective over a large range of sizes.

Rotors of the Beams type are employed advantageously when the geometry of the rotor is such that the moment of inertia in the plane of rotation is large as is pointed out by the authors of this paper. However, when this is inconvenient or perhaps impossible and when the other advantages of a constrained journal running in fixed bearings are important considerations, then the design described herewith is called for.

While it may appear that possible applications for this type of bearing operation are limited, a number of cases which should bear investigation come readily to mind. Most small very high-speed devices are obviously in this category, such as small spray nozzles, high-speed grinding wheels for small internal grinding, very small drills, gyroscopes and microtomes, to name a few in addition to the centrifuge and optical use already mentioned.

[5] Consulting Engineer, Litchfield, Conn. Mem. ASME.

[6] U. S. Patent No. 2,603,539.

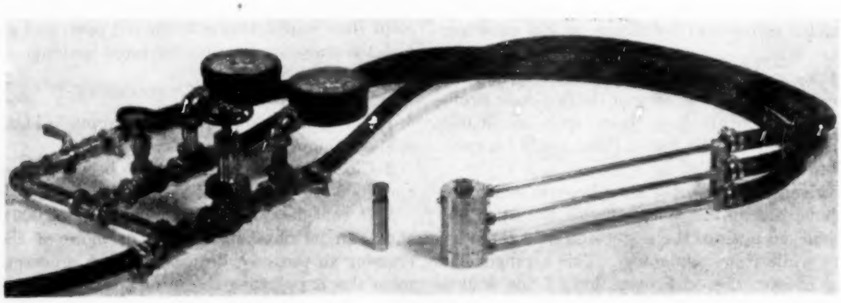

Fig. 6 240,000 Rpm Air Motor

In addition to these more obvious applications the writer is foolhardy enough to believe that someday we may find it entirely feasible to apply the principles of gas-film lubrication to such things as steam turbines, gas turbines, or other large units where the load is steady and rubbing speed is high. The use of gas lubrication in situations where the bearing is too hot for ordinary lubricants is immediately suggested. There should be some concern in this country with the imagination and courage to explore this nearly virgin field.

J. A. Frost.[10] The writer has used a spinner of the type described in the paper with great success.

Because of the high speed required in testing a particular device, it was necessary to make certain modifications over the one described. These included a smaller stator and rotor and elimination of the extra apparatus not pertinent to the problem at hand.

Using a $2^{1}/_{2}$-in-diam rotor and 100 psi air pressure, 66,000 rpm was obtained. The device to be tested was mounted in a cavity in the center of the rotor. Operation of the device was observed through the use of a strobotac and speed was checked, using a photoelectric cell of the type described.

The writer agrees with the authors' conclusion and highly recommends the air spinner for laboratory use.

H. E. Ruehlemann.[11] The writer has designed and operated air spinners for the past 10 years, and a spinner similar to the one described in the paper has been used by the Ammunition Division of the Naval Ordnance Laboratory for the past 2 years. This type of spinner, with minor modifications to meet our special needs, has performed satisfactorily in all respects. The spinner is dependable, has no maintenance problem, and is ideally suited for laboratory testing of components, and multiple-purpose developmental apparatus, at various spin rates. The ability of the spinner to operate properly with reasonable unbalance is truly a desirable characteristic in laboratory testing.

It is felt that the claims made for the spinner when operated under the conditions specified are truly indicative of the versatility of this device.

[10] Mechanical Engineer, Ammunition Division, U. S. Naval Ordnance Department, White Oak, Md.

[11] Project Engineer, Ammunition Division, U. S. Naval Ordnance Laboratory, White Oak, Md.

Authors' Closure

The comments of Mr. Brewster, Mr. Frost, and Mr. Ruehlemann are greatly appreciated.

It is hoped that this paper has helped to make a little more widely known the characteristics of air-supported and air-driven spinners. We feel that such a spinner is a most important tool in a development laboratory concerned with high-speed rotation of small devices.

Since this paper was written, some consideration has been given to a redesign to make the air spinner suitable for production-line testing of small timing mechanisms. At present, testing is time-consuming and not particularly well suited to mass-production testing. About one minute is needed for installation of the device to be tested, three minutes to accelerate to 36,000 rpm, and one-half minute to bring the spinner to a stop after the test is completed. In an attempt to speed up this process, the following possibilities are being considered.

1 Faster installation, of the device to be tested, by use of snap-action locking detents rather than the present two hold-down screws.

2 A hand or foot-lever action for bringing the auxiliary-equipment assembly down over the rotor and stator. This action would also initiate the air drive supply.

3 A feedback system to hold automatically the proper speed when reached and initiate the monitoring operation.

4 An air brake consisting of reverse jets and actuated by completion of test operation.

5 Use of a bank of such spinners side by side—enough to keep an operator continually busy. The operator's complete job would be inserting the test device, initiating the operating sequence, removing the test device, and recording the test results.

A Method of Obtaining Shear Stress-Strain Graphs by Interpretation of Moment-Twist Data

By P. H. KAAR,[1] CHICAGO, ILL.

Many requests are made by industry for information from the Fritz Engineering Laboratory at Lehigh University. Among these was one from a manufacturer wishing to increase the strength of his industrial truck axles. The manufacturer was considering several different steels and wanted comparative strength data. The information was compiled by use of shear stress-strain curves. The method described here to obtain the shear stress-strain curves is not new, but is not often used. The procedure is simple to apply and merits the attention of others concerned with comparative strength and shear properties of axial round shafts.

Nomenclature

The following nomenclature is used in the paper:

a = radius of shaft, in.
I_p = shaft polar moment of inertia, in.^{4}
M = applied torque or moment, in.-lb
θ = angles of twist of axle or shaft per unit length, radians per in.
τ = unit shear stress, psi
γ = unit shear strain, in. per in.

Theory

If, for any round uniform section of isotropic material, the moment-twist curve $M = F(\theta)$ is known, Fig. 1, the shear stress-strain curve can be found in the following manner:

The unit shearing strain of the bar depends on the shearing stress τ and $\tau = f(\gamma)$ is a continuously increasing function of the unit shearing strain, γ. $\tau = f(\gamma)$ is the stress-strain curve for the material for shear

$$M = \frac{2\pi}{\theta^3} \int_0^{\gamma_a} f(\gamma)\, \gamma^2 d\gamma \qquad [1]^2$$

or

$$M\theta^3 = 2\pi \int_0^{\gamma_a} f(\gamma)\, \gamma^2 d\gamma \qquad [2]$$

$$\gamma_a = a\theta$$

differentiating Equation [2] with respect to θ

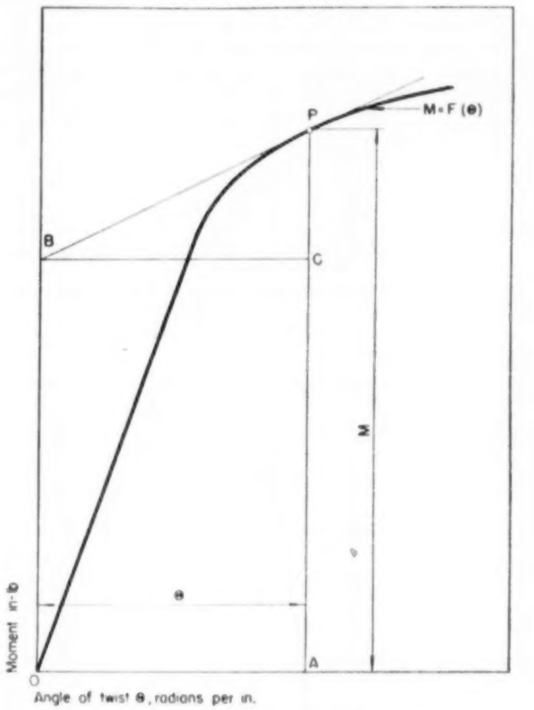

Fig. 1 Moment-Angle of Twist Curve

$$\frac{d}{d\theta}(M\theta^3) = 2\pi\, f(a\theta)\, a^3\theta^2 = 2\pi a^3\theta^2\tau_a \qquad [3]$$

solving for τ_a

$$\tau_a = \frac{1}{2\pi a^3\theta^2} \frac{d}{d\theta}(M\theta^3) \qquad [4]$$

since $$\frac{1}{\theta^2}\frac{d}{d\theta}(M\theta^3) = \theta\frac{dM}{d\theta} + 3M = CP + 3\,AP \qquad [5]$$

$$\tau_a = \frac{1}{2\pi a^3}(CP + 3\,AP) \qquad [6]$$

Description

The manufacturer furnished two axle specimens $1^3/_4$ in. diam, 28 in. long, of 4340 and 1045 steel which were tested in torsion. The axle of 4340 steel was induction-hardened to a depth of $^1/_8$ in. and had a hardness of Rockwell 58, C scale. The axle of 1045 steel was heat-treated to a hardness of Rockwell 35, C scale.

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[1] Research Engineer, Armour Research Foundation of Illinois Institute of Technology, Chicago, Ill.; formerly, Engineer of Tests, Assistant Professor, Fritz Engineering Laboratory, Lehigh University, Bethlehem, Pa.

[2] For the complete development of this expression see "Theory of Flow and Fracture of Solids," by A. Nadai, second edition, McGraw-Hill Book Company, Inc., New York, N. Y., vol. 1, 1950, chapter 21, pp. 347–349.

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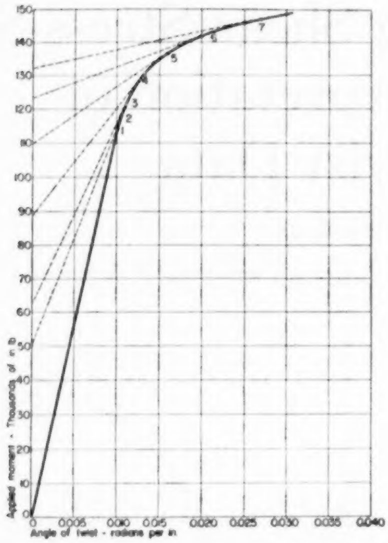

Fig. 2 Moment-Angle of Twist Curve 4340 Steel Axle
(Diam = 1.754 in. I_p = 0.929 in.[4])

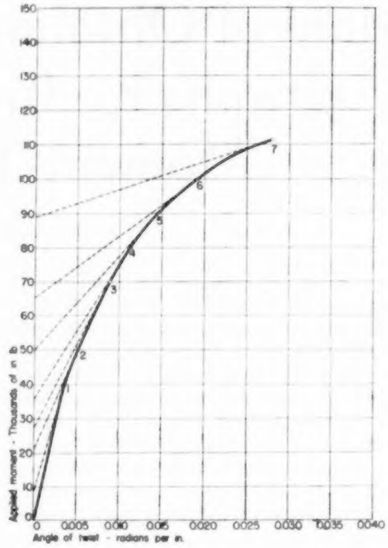

Fig. 3 Moment-Angle of Twist Curve 1045 Steel Axle
(Diam = 1.753 in. I_p = 0.927 in.[4])

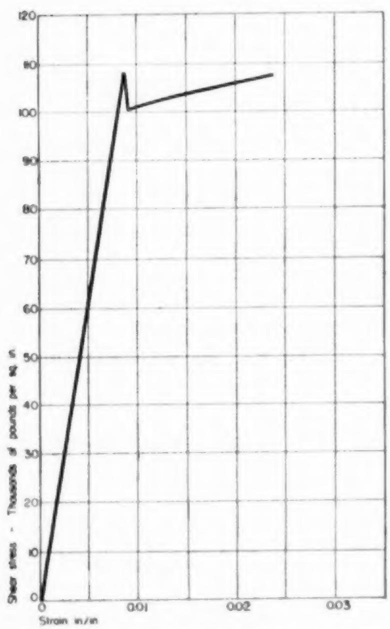

Fig. 4 Shear Stress-Strain Curve 4340 Steel Axle
(Diam = 1.754 in. I_p = 0.929 in.[4])

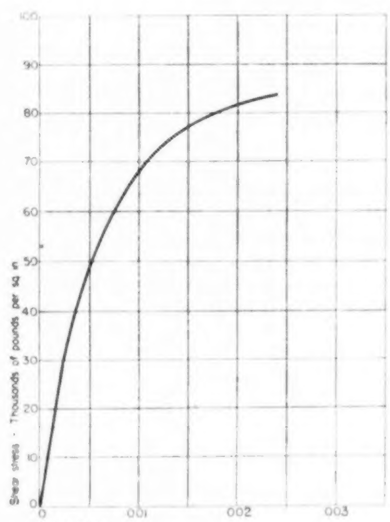

Fig. 5 Shear Stress-Strain Curve 1045 Steel Axle
(Diam = 1.753 in. I_p = 0.927 in.[4])

Moment was applied by the 2,000,000-in-lb-capacity torsion testing machine at the Fritz Engineering Laboratory.[8] The angle of twist was determined by measuring over a 10 in. gage length. A surveyor's level bubble tube was mounted on a bar, one end of which contained a micrometer barrel perpendicular to a .d extending through the bar. As the specimen was twisted, the angular-displacement tangent was determined by the distance the micrometer point had to be advanced to return the bubble to center. Seats for this leveling instrument were provided by attaching collars with flat bars to the axle. The bars projected at right angles to the specimen and rotated with it during the test. Dual readings were taken—one reading on each side of the axle—for each applied moment.

Figs. 2 and 3 show the moment-twist curves obtained by these

[8] "A Torsion Testing Machine of 2,000,000 Inch-Pounds Capacity," by F. K. Chang, et al., ASTM Bulletin No. 160, September, 1949.

TABLE 1

Point	Radians per in.	Strain $a(\theta)$ in/in	CP, in-lb	$3(AP)$, in-lb	$CP + 3(AP)$, in-lb	$\frac{1}{2\pi a^3}[CP + 3(AP)] =$ shear stress, psi
			For 4340 Steel Axle			
1	0.0100	0.0088	114600	343800	458400	108200
2	0.0104	0.0091	68700	355500	424200	100100
3	0.0110	0.0096	59800	365400	425200	100300
4	0.0132	0.0116	42100	392400	434500	102500
5	0.0160	0.0140	27100	411000	438100	103400
6	0.0200	0.0175	18700	426000	444700	104900
7	0.0270	0.0237	15150	441900	457050	107900
			For 1045 Steel Axle			
1	0.0036	0.0031	30800	120000	150800	35600
2	0.0050	0.0044	28500	150000	178500	42200
3	0.0088	0.0077	42200	210000	252200	59600
4	0.0112	0.0098	44000	240000	284000	67100
5	0.0143	0.0125	40000	270000	310000	73300
6	0.0188	0.0165	34400	300000	334400	79000
7	0.0276	0.0242	22000	333600	355600	84000

tests. In each case seven tangents have been constructed to the curve to illustrate the graphical interpretation. In Fig. 2 the first tangent drawn coincides with the moment-twist curve itself—being drawn at the top of the elastic portion of the curve. By applying the theory previously developed, and using the notations defined in Fig. 1, the data in Table 1 were developed.

Figs. 4 and 5 show the shear stress-strain curves obtained by this graphical interpretation. The curve shown in Fig. 4 is of particular interest because it shows both an upper and lower yield point. Shear stress-strain curves produced from moment-twist curves having a sharp bend just beyond the elastic range of the material show this double yield property.

While separate stress computations are made frequently after observing applied moment, angle of twist, and diameter of shaft, the whole shear stress-strain curve from these data is seldom plotted. This plotting, easily done by the described method, is of great aid in the selection of axle steels.

Corrosion and Erosion in the Synthetic-Fuels Demonstration Plant

By G. D. GARDNER[1] and J. T. DONOVAN,[2] LOUISIANA, MO.

Corrosion and erosion during 2 years of intermittent operation of the first American coal-hydrogenation plant are described as well as solutions and improvements developed. Isolated cases of localized corrosion fatigue, caustic embrittlement, and pitting are described. Proper alloy-steel selection has minimized the effect of hydrogen attack in high-pressure equipment except where design temperature has been exceeded. High-pressure lines and valves handling oil-solid mixtures containing absorbed gases have evidenced greatest erosion rates.

Introduction

Prior papers[3,4,5] describe process equipment, selection of materials, and early metal failures incident to the construction testing, and initial operating phases of the hydrogenation demonstration plant.

Improved design and fabrication techniques have eliminated most of these earlier difficulties, while the original selections of material have proved to be sound, requiring little alteration.

Inspection of selected equipment and process piping after each liquid-phase and vapor-phase run has disclosed very little corrosion in the high-pressure equipment and only nominal corrosion in most low-pressure equipment.

Stress corrosion cracking of high-pressure instrument tubing has caused considerable concern, while corrosion on the water side of low-pressure tube-and-shell coolers has required constant surveillance. Hydrogen-sulphide corrosion has occurred in tube-and-shell exchangers, let-down drums, and accumulators, where temperatures are below 300 F. Three tanks handling caustic have failed owing to caustic embrittlement of the heat-affected zone near welds that were not stress-relieved. Corrosion of a minor nature also has been traced to nitrous acid formed in the inert-gas producer, and pump plungers and valve stems of 12 per cent chromium steels have pitted rapidly during shutdown periods.

There have been a few cases of severe erosion. The valves throttling the "heavy-oil let-down" (HOLD), a mixture of oil, gas, and solids, have eroded rapidly, although the most abrasion-resistant materials obtainable have been used. High-pressure fittings and equipment following these valves also have been eroded wherever changes in direction occur.

By providing parallel lines and multiple-valve manifolds where erosion was anticipated, plant operation has never been interrupted. Changes in design have reduced or isolated the erosion to readily replaceable parts.

Corrosion

Stress Corrosion Cracking of High-Pressure Instrument Tubing. All high-pressure instrument tubing in the hydrogenation demonstration plant connecting the process lines with remote controlling, recording, and indicating instruments is made of titanium-stabilized stainless steel (AISI Type 321). The normal operating pressure for this $^9/_{16}$-in-OD × $^3/_{16}$-in-ID tubing is 10,000 psi, and temperatures range from atmospheric to 300 F. Fluids in the line may be any of those incident to the process, including hydrogen, hydrocarbon gases, hydrocarbon liquids, pastes of oil and coal, and tar acids.

During initial operation of the plant, leaks occurred in the instrument tubing, although none had been found during preliminary pressure tests above the operating pressure. These leaks occurred at circumferential and longitudinal cracks, Fig. 1. Many small microvoids were found on the inner surface of sections cut from the ruptured tubes and also in new and other used tubing which had not ruptured, Fig. 2. Some of the cracks were typical of those found in sensitized stainless steel that has corroded intergranularly, Fig. 2(*a*). Others were transgranular, Fig. 2(*b*), a form typical of stress-corrosion cracks. A check analysis of a sample of the tubing revealed that the titanium was only slightly below the normal range for the carbon content of the steel. Samples taken adjacent to the cracks were boiled in cupric-sulphate solution with and without a special sensitization treatment at 1200 F. Neither treatment embrittled the tubing; therefore it was considered to be stabilized satisfactorily.

Although there were voids in the tubing and it was subjected to stress, the normal process fluids did not contain a suitable corrodent for stress corrosion cracking. Investigation, however, disclosed small amounts of hydrochloric acid in adjacent lines that could have become concentrated in these static-tubing sections. This acid was found to have come from carbon tetrachloride which had not been flushed completely from the system following the cleaning procedure. Under the influence of pressure and temperature in the presence of iron oxide that would act as a catalyst, the carbon tetrachloride was decomposed into carbon and free chlorine which combined with the water to form the acid. Stress concentration at the microvoids in the presence of hydrochloric acid resulted in stress corrosion cracking of the tubing.

The acid was flushed out of the installed tubing, which was then tested hydrostatically at $2^1/_2$ times the operating pressure. No failure has occurred in tubing thus cleaned and tested. As chlorides are seldom present in the process streams, the presence of microvoids in the remaining tubing is not considered a serious hazard.

However, since these voids are undesirable in high-pressure tubing, their cause was investigated. It was found that tool-

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[3] "High-Pressure (10,300 Psi) Piping, Flanged Joints, Fittings, and Valves for Coal-Hydrogenation Service," by J. H. Sandaker, J. A. Markovits, and K. B. Bredtschneider, Trans. ASME, vol. 72, 1950, pp. 365–372.

[4] "High-Pressure Vessels in Coal-Hydrogenation Service," by J. T. Donovan, M. Josenhans, and J. A. Markovits, Trans. ASME, vol. 72, 1950, pp. 357–363.

[5] "Metallurgical and Fabrication Considerations in the Coal-Hydrogenation Demonstration-Plant Construction," by B. H. Leonard, Jr., G. D. Gardner, and J. A. Markovits, Trans. ASME, vol. 72, 1950, pp. 379–383.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, June 10, 1952. Paper No. 52—F-28.

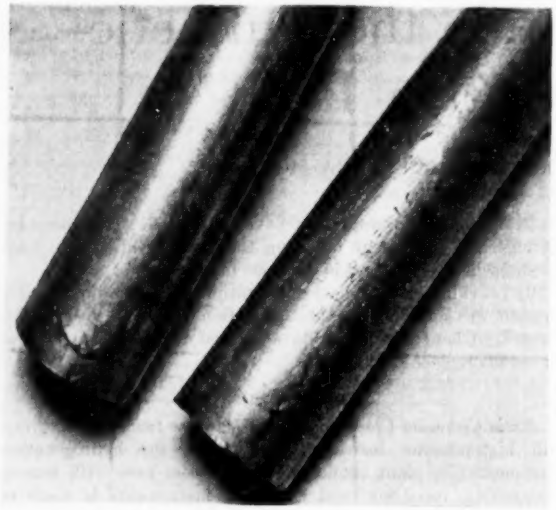


FIG. 1 STRESS-CORROSION CRACKS IN HIGH-PRESSURE INSTRUMENT TUBING

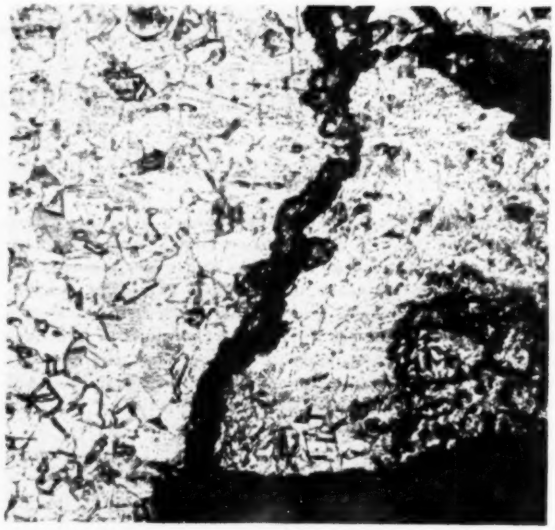


FIG. 2(*a*) LONGITUDINAL CRACK IN INSTRUMENT TUBING
(Etchant, aqua regia; ×100.)

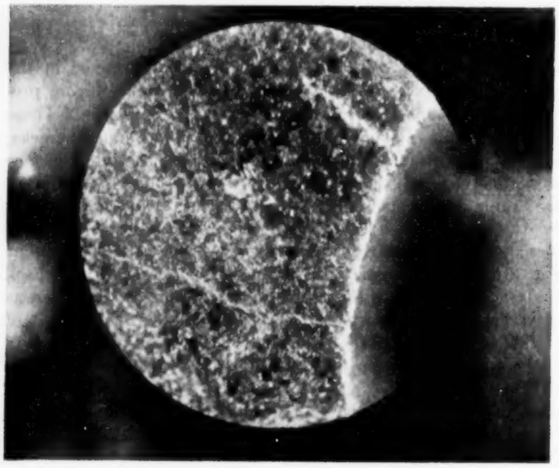


FIG. 2 VOIDS IN HIGH-PRESSURE INSTRUMENT TUBING
(Etchant, aqua regia; ×100.)

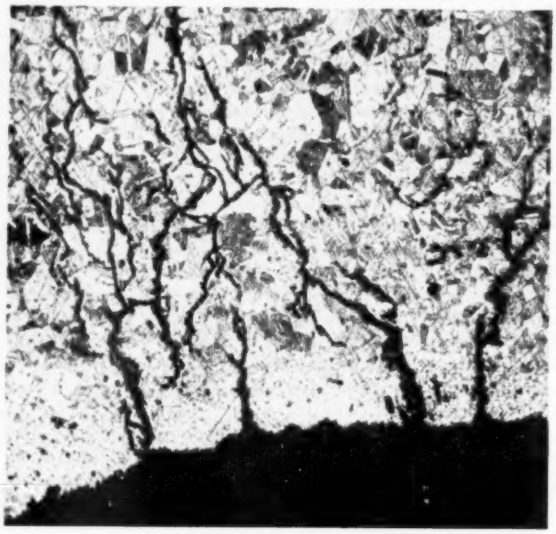


FIG. 2(*b*) RADIAL CRACK IN TUBING
(Etchant, aqua regia; ×100.)

marks, scratches, and laps are always likely in certain stages of tubing production. These marks are increased in depth during subsequent size-reducing steps and remain as stress raisers unless removed.

On the basis of this investigation, a new tubing specification was prepared. It included the following requirements:

1 During the drawing-down process, the tubes shall be internally ground at the smallest diameter practical, but in no case greater than 1 in. ID.

2 This internal preparation shall be done in a manner to remove all internal cracks, seams, scratches, mandrel marks, or other defects resulting from prior processing.

3 Further size reduction shall be made in not less than six steps with an intermediate anneal at 2000 F after each cold reduction.

4 A sample shall be cut from each length of tubing for microscopic examination. At ×100 there shall be no evidence of cracks or seams on inside surface. Well-rounded voids 0.005 in. deep will be satisfactory.

Corrosion Fatigue of Bourdon Tubes in Pressure-Indicating Equipment. The discovery of hydrochloric acid in instrument tubing, as just mentioned, gave further insight into the cause of the rapid failure of the Bourdon tubes reported in a previous paper.[4] Hydrochloric acid also might have been a factor in the subsequent failure of the welded socket joint of several Bourdon tubes which had been designed for 20,000 psi. The welds in question had slag inclusions which acted as stress raisers in accelerating failure of the tubes. The importance of high-quality welding on parts subjected to cyclic stress cannot be emphasized

too strongly. Since the installation of tubes made of AISI Type 316 stainless steel designed for 20,000 psi, there have been no failures of the tubes in over 5000 hr of operation. The hysteresis of this type material has not been objectionable in either indicating or control instruments.

Pitting of 12 Per Cent Chromium-Steel Plungers and Valve Stems. The packing for the original high-pressure injection pumps was held in place by bronze support rings, and the plungers were a hardened 12 per cent chromium steel. After preliminary pumping tests were made with water, some of the pumps stood idle for several months causing a ring of deep pits in several of the rods where they had been in contact with the bronze rings, Fig. 3. This galvanic pitting has been eliminated by operating the pumps periodically during idle periods to recoat the plunger with oil.

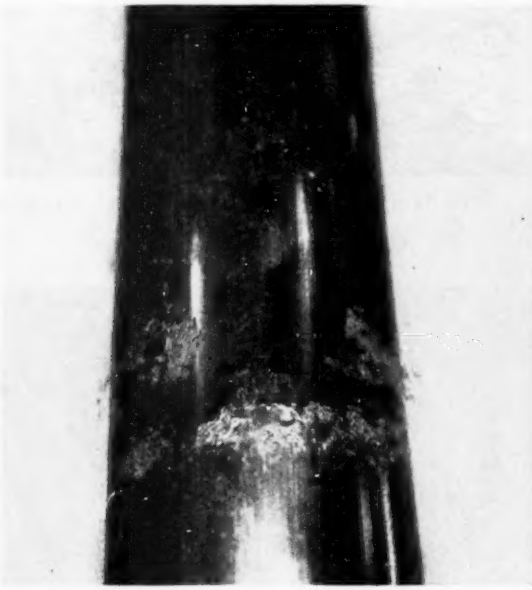


Fig. 3 Pits in 12 Per Cent Chrome-Steel Pump Plunger

Similar-type pitting has occurred on 12 per cent chromium-steel valve stems that have operated through a braided packing containing copper. This galvanic corrosion occurred where moisture had accumulated on the stems during idle time, either from atmospheric moisture or from water remaining after hydrostatic tests. The substitution of aluminum or lead-base packing, combined with periodic operation to lubricate the parts, has eliminated pitting.

Nitrous Acid in Inert-Gas Compression Lines. All high-pressure piping and equipment in the high-pressure plant are pressure-tested with inert gas before each run. During the final stages of plant construction and assembly, multiple tests were necessary on both equipment and piping.

The throttling valves in the high-pressure compressor manifolds were eroded rapidly during compression of this gas to 10,000 psi. The same valves had shown no evidence of erosion after much longer periods in hydrogen compression to even higher pressures. Therefore it was assumed that some constituent of the inert gas was responsible for this rapid wear. The plugs and seats of these valves also were deeply corroded, although little corrosion appeared elsewhere in the system.

Small concentrations of nitrous oxide were found in the gas as it left the inert-gas generator. During compression and cooling, water condensed, and at the intermediate pressures, combined with the nitrous oxide to form nitrous acid. It was concluded that the wear on the valve parts was corrosion-erosion resulting from the combined effects of nitrous acid and the high velocity through the throttling-valve port.

Steam has been added in the firebox to lower the temperature which has cut down the formation of nitrous oxide. Copper baffles placed in the outlet line from the generator have accelerated the decomposition of any nitrous oxide formed. These two corrective steps virtually have eliminated the nitrous acid, and the valve life has been increased indefinitely.

Hydrogen Attack. Piping and equipment specifications for 10,000-psi operating pressure and 8000-psi hydrogen partial pressure were divided into three general classifications as follows:

Temperature	Material
0– 375 F	SAE 1040
	AISI 4130
375– 850 F	Croloy 9M (9% Cr, $^1/_2$% Mo)
850–1000 F	AISI 316

The 9 per cent chromium steel has good resistance to hydrogen attack at all temperatures. Above 850 F, however, the creep strength of this alloy is not adequate for the intended application, and the austenitic stainless steel AISI Type 316 was chosen.

Where operating temperatures have not exceeded the design temperature by more than 100 F, there has been no detectable hydrogen attack after more than 5000 hr of operation. One section of carbon-steel tubing, which operated at 600 F for several hundred hours, had barely discernible edge decarburization. A section of 1 per cent chromium tubing, which had operated at 600 to 900 F for 150 hr, had been decarburized to a depth of $^1/_{16}$ in.

Sample specimens of various alloys, including the piping materials listed, straight chromium-alloy steels ranging from 2 to 20 per cent chromium, and various austenitic stainless steels, have been exposed to operating conditions in the preheaters and the converters. Many of the samples were furnished by Babcock & Wilcox Tube Company, under a co-operative agreement for the study of the effect of temperature, stress, and hydrogen on certain straight chromium steels. Others were furnished by Carpenter Steel Company and Armco Steel Company. Only limited hydrogen embrittlement has been found in any except the carbon and lowest-alloy steel materials to date. There has been some embrittlement of ferritic stainless steels, but it appears to have been due to excessive temperatures which occurred during runaway reactions when temperatures in some locations exceeded 1800 F for short periods. Austenitic stainless steels in some cases were embrittled slightly, possibly as a result of sensitization during the runaway reactions.

A bellows-type lens-ring gasket, which had been in service for 1000 hr or more at 10,000 psi and 900 F, cracked open during a hydrostatic test at 15,000 psi, Figs. 4 and 5. The test reports for this material indicated that it was AISI Type 405 with a chromium equivalent of 9.57, based on Thieleman's factors.[6] This would indicate that exposure to the noted temperatures and pressures should not have resulted in embrittlement.

Microscopic examination of a section of the gasket revealed a deep etching case in which microfissures and cracks were found, Fig. 6. A similar zone was noted in the area immediately surrounding the cracks which extended through the wall of the gasket. A second adjacent zone, Fig. 7, had a higher percentage of carbides precipitated within the grains than is normally expected, and there was a third zone, Fig. 8, near the interior of the gasket which was practically free of precipitated carbides. This

[6] "Some Effects of Composition and Heat-Treatment on the High-Temperature Rupture Properties of Ferrous Alloys," by R. H. Thieleman, Trans. ASTM, vol. 40, 1940, pp. 788–804.

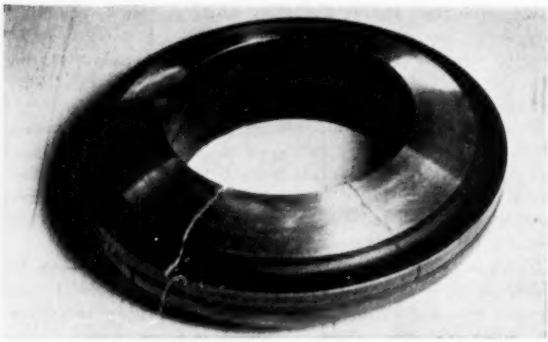


Fig. 4 Cracked Lens-Ring Gasket Made From AISI Type 430 Stainless Steel

Fig. 5 Cracked Lens-Ring Gasket Broken at Cracks to Show Fractured Surface

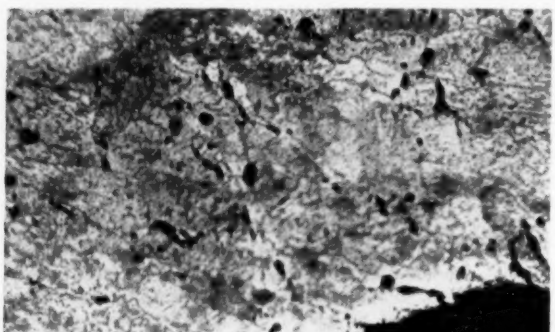


Fig. 6 Inner Edge of Stainless Steel—AISI Type 430—Lens-Ring Gasket

(Etchant, picric acid in HCl; ×100.)

structure and the large grain size were not expected in Type 405 stainless steel, and therefore samples were submitted to the A. O. Smith Corporation and to Allegheny Ludlum Steel Corporation. They reported that the material analysis conformed to that of AISI, Type 430 stainless steel rather than Type 405

Fig. 7 Zone Near Edge of Gasket Having High Concentration of Carbide Particles

(Etchant, picric acid in HCl; ×100.)

Fig. 8 Zone With Lower Concentration of Carbide Particles

(Etchant, picric acid in HCl; ×100.)

as shown in the test reports. On this basis, it was concluded that the gasket had been embrittled by heating in the embrittling temperature range of this type steel. Other gaskets made of the same material were similarly embrittled after indefinite periods in service.

Pitting of Underground Lines. A unit of the ammonia plant, now supplying purified compressed hydrogen, began operations in November, 1942, and was shut down in August, 1945. After about 15 months in stand-by condition, it was reactivated in 1946, and operated for 5 months.

During this latter operation, holes and pitting were discovered in one of the underground water lines. An increasing number of underground lines have been replaced each year because of pitting

corrosion. All of these lines had been coated with a bitumastic coating for corrosion protection, but had not been paper-wrapped.

The first line that leaked was laid in a clay soil containing a small amount of broken stone. Some sections of the coating on the pipe showed mechanical damage. It was concluded that the pits which finally corroded through the pipe were initiated at breaks in the coating. Galvanic-type concentration-cell corrosion caused preferential attack at the deepest part of these pits, continuing until the pipe was perforated.

Wherever possible, new lines have been run above ground or in concrete-lined trenches. Where lines have been repaired or replaced, the coating has been checked carefully with high-frequency, spark-type detectors to be sure that there were no initial breaks in the coating. A cathodic-type system is being considered as a means of extending the life of all underground lines.

Stress Corrosion Cracking of Caustic Storage Tanks. Three tanks of $^3/_{16}$-in. carbon-steel plate used for storing caustic developed cracks adjacent to the welds after very short service. One tank had been used for storing 10 per cent caustic for 15 months and then held 50 per cent caustic for 35 days at temperatures of 200 to 300 F. Several cracks were found at the point where the nozzle-reinforcing plates had been welded to the tank shell, Fig. 9. These cracks were repaired by chipping and welding, and the tank was returned to service but developed additional cracks at similar points after 6 days. Cracks were also found where the gage-board support had been welded to the tanks, Fig. 10.

A second tank had been used for storing 10 per cent caustic for 8 months and 50 per cent caustic for 3 months at 200 to 300 F before failure, adjacent and parallel to the girth weld.

A third tank containing 50 per cent caustic for 4 months developed cracks across the vertical weld of the shell at the caustic level.

These failures are examples of stress corrosion cracking (caustic embrittlement) in which unrelieved welding stresses accelerate failure.

Intergranular Corrosion. Several stainless-steel thermowells were used in the activated-charcoal purification vessel. In this vessel, sulphur compounds are removed from the gas stream passing through the bed. The charcoal is reactivated by passing superheated steam back through the bed, whereby water solutions of sulphur compounds are formed. The thermowells had been welded to a carbon-steel flange for connection to the vessel. After approximately 500 hr of operation, interspersed with several months of idle time, the stainless-steel tubes developed cracks near the weld which were traced to intergranular corrosion, Fig. 11. The acid solution, which condensed on the cooler thermowell, initiated intergranular corrosion of the sensitized area near the weld. It also deeply pitted the ends of the thermowells. New thermowells are to be made of stabilized austenitic stainless steel.

Fig. 9 Crack in Caustic Tank Near Nozzle Weld

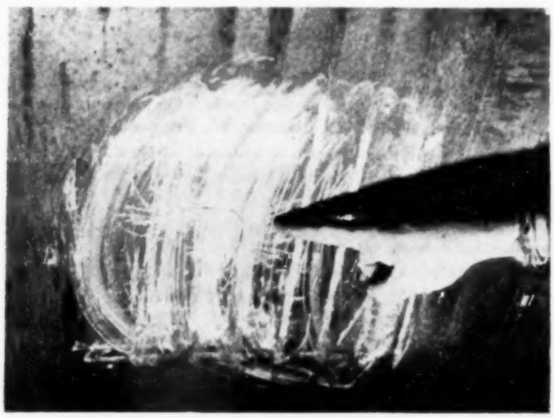


Fig. 10 Crack on Inside of Tank Wall Where Gage-Board Bracket Had Been Welded on Outside of Tank

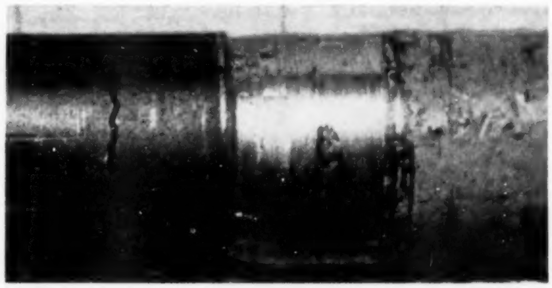


Fig. 11 Thermowell Showing Intergranular Corrosion Crack Near Weld

Erosion

Heavy-Oil Let-Down Throttling Valves. The severe throttling valves, which throttled the HOLD containing abrasive solids and absorbed gases from 10,000 psi to atmospheric pressure, have been eroded most rapidly. These valves have an orifice diameter of $^1/_8$ in., and were designed to pass a flow of 500 to 800 gph. This mixture normally contains approximately 10 to 20 per cent abrasive solids, and 3 per cent by volume of dissolved gases at 700 atm. Inner valves made of tungsten-titanium carbide were eroded beyond further use in 8 to 24 hr while controlling the flow to 300 gph. At this rate there was close clearance between the plug and the seat, resulting in high velocity through the small annular area. A secondary restricting orifice placed downstream from the valve absorbed only a small percentage of the total pressure drop. When the rate was increased to 600 gph, the restricting orifice absorbed a larger portion of

Fig. 12 Eroded Cemented Tungsten-Titanium-Carbide Seat Insert and Plug for Severe Throttling Control Valve

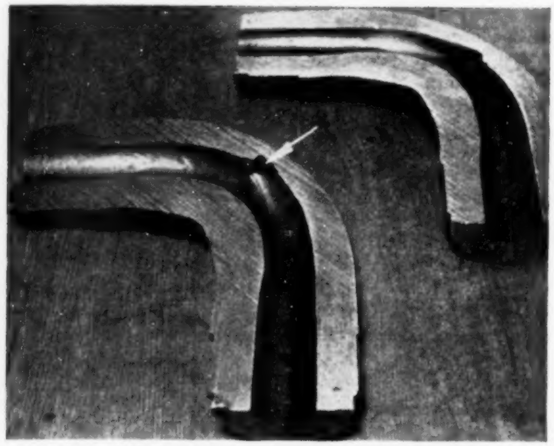


Fig. 14 Eroded 1-In. High-Pressure Elbow

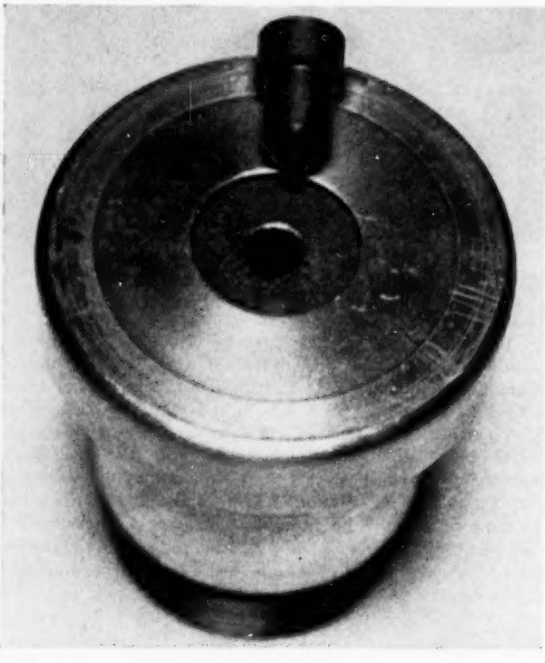


Fig. 13 Eroded Cast Boron-Carbide Seat Insert and Plug for Severe Throttling Control Valve

the total pressure drop. Velocity across the valve with the higher plug lift was consequently less, extending valve life to 3 days, Fig. 12. Cast boron carbide has given considerably longer service lasting 15 days under similar operating conditions, Fig. 13.

Close sizing of the orifice to fit the design flow rates is important. Too small an orifice will not pass the required volume, and one too large will not take an appreciable proportion of the pressure drop.

HOLD Targets and Fittings. The HOLD was carried from the control valves to the let-down vessels in 1-in. high-pressure lines. Ahead of each vessel was a restricting orifice followed by a drilled target, primarily intended to restrict the flow if the control valves became inoperable, and also to act as a secondary pressure-reducing point, thereby decreasing the severity of the valve service.

After approximately 5 weeks of operation, a hole was cut through the heavy-wall elbows on each of the parallel lines running to the let-down vessel, Fig. 14. To combat this high erosion rate, a larger-diameter tee containing a replaceable wear plug was installed in place of the elbow.

The hard-faced plug target has absorbed the energy satisfactorily and the erosion that previously had been concentrated on the elbow. Fig. 15 shows the arrangement of the plug target following the restricting orifice. Fig. 16 shows the magnitude of the erosion on two of the special plug targets after 6 weeks of service.

Other Control Valves. The plugs and seats of control valves handling relatively clean liquids and gases were not eroded as rapidly as those handling HOLD. Tungsten-titanium carbide has been the most satisfactory material where cross-sectional area of the parts was greater than 0.02 sq in. Those parts with smaller cross-sectional area have been made of stellite No. 1 with good results. This material is less abrasion-resistant, but also less brittle.

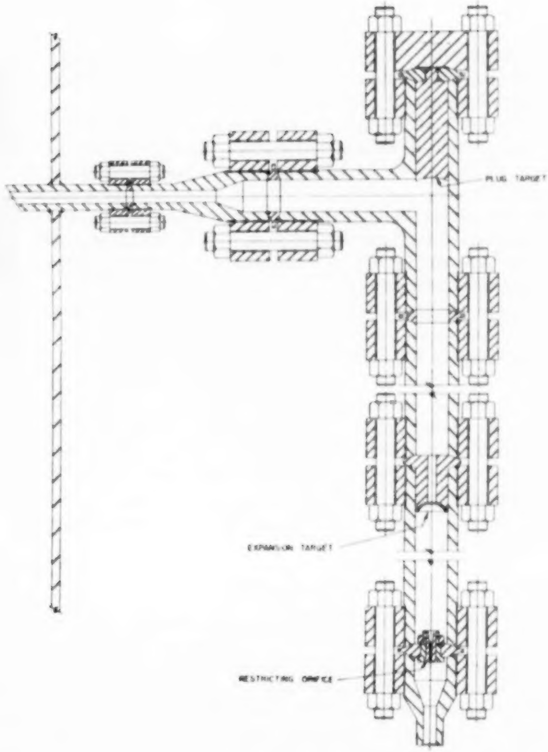
Hand-Operated Valves. Tungsten-titanium-carbide plugs and seats in hand-operated throttling valves have given good service. The ring-type seat insert has been satisfactory except on the $^3/_8$-in. valve where the cross-sectional area was so small that the seats cracked as a result of closing forces. On less severe service, stellite seats have proved satisfactory in all respects.

When throttling valves are operated at approximately closed position, the high flow velocities have eroded the steel body below the insert, Fig. 17.

The seating surfaces of hand-operated block valves faced with stellite have been satisfactory, except when hard particles of abrasive material lodge on the seat, preventing complete closure. When these valves are not closed completely, the seat and the body behind the seat are eroded rapidly. Very fine cracks on the stellite have been found, and in some cases have initiated erosion. With upward flow, the body of the valve becomes eroded, together with the seat; downward flow erodes the replaceable body seat ring.

Pump Valves. The most severe erosion of valves in high-pressure injection pumps has been in coal-paste service. The

Fig. 15 Restricting Orifice and Target Assembly



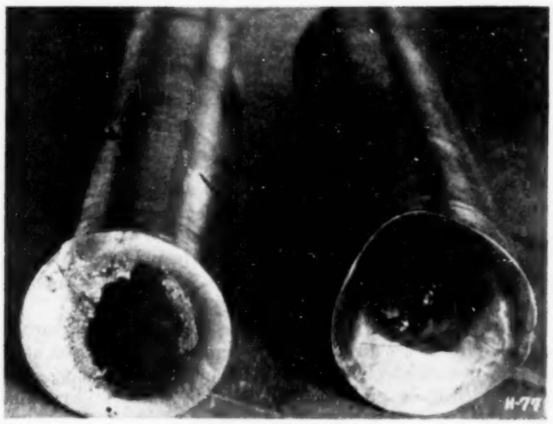


Fig. 16 Erosion of Plug Target

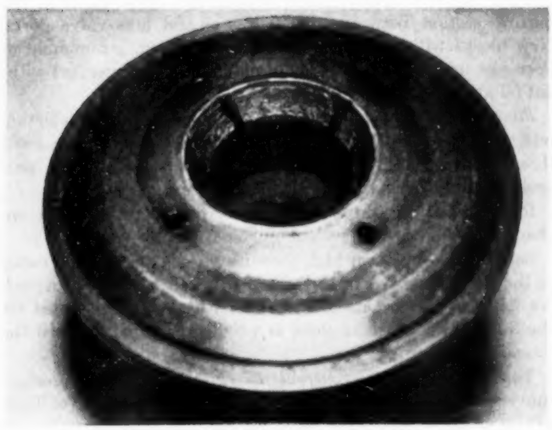


Fig. 18 Erosion of Stellited Pump-Valve Seat

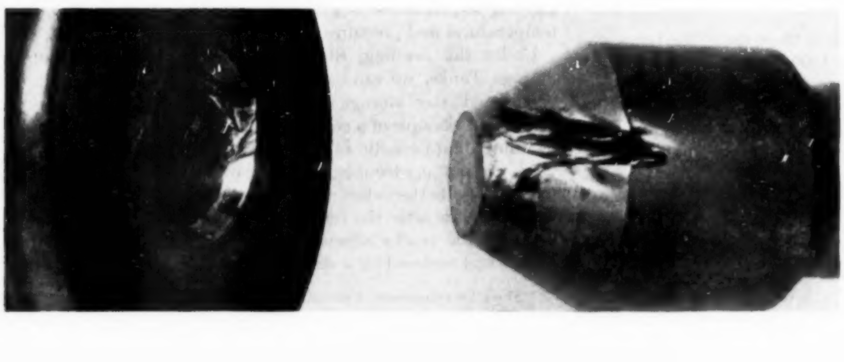


Fig. 17 Erosion of Valve Body and Stem

valves in the original pumps[7] were mushroom type, with large-radius valve face and conical seats of AISI Type 440-C stainless steel. These were eroded after a few hours of service.

Balls 2 in. diam of the same material hardened to Rockwell C-56 on conical stellited seats gave 2 weeks of service. With this type valve, line contact is achieved, and hard particles are either cut or washed from the surface. Cracks have developed occasionally at the terminal junction of the welded stellite seats causing erosion. In others, the stellite has deformed to the shape of the ball, offering wider contact surface for particle build-up initiating erosion, Fig. 18.

Other materials tested include a 12 per cent chrome ball valve with graphitic tungsten tool-steel valve seats hardened to 52 RC. After approximately 360 hours of service on water and clean oil, the graphitic-steel seat was eroded very deeply in one narrow zone. In the same service an alternate chromium-vanadium steel seat was eroded lightly at many points. Spring-loaded, mushroom-type, high-chrome valves with a 45-deg conical face and seat have proved more satisfactory in clean-liquid service.

Cast uniform-hardness ring-type stellite inserts with $^3/_{32}$-in. contact surface have been prepared for the next liquid-phase run for paste service.

Other Metal Failures

Fatigue of Pump Blocks. The high-pressure injection pumps, as originally designed, had many sharp edges at the intersection of the valve ports with the main cylinder bore.[7] There were also sharp edges where transverse holes intersected the plunger bore. These transverse holes were closed with threaded and seal-welded plugs. After relatively short service, the seal welds cracked and could not be repaired satisfactorily.

After 300,000 stress cycles, cracks also developed at the narrow section between the discharge and inlet-valve ports. New blocks with smooth internal surface finish, a minimum of openings, and fillet radii of $^1/_8$ in. minimum have operated without failure for over 3,000,000 cycles.

Fatigue. Steam-cylinder rods on the high-pressure injection pumps had been designed with sharp changes of section near the threads. After approximately 900,000 cycles one of these rods broke, through fatigue at the root of the thread.

Replacement rods provided with generous fillet radii at section changes have withstood several million cycles.

Some flange bolts on high-pressure vessels and pumps broke at the root of the last thread where no relief had been provided. No failures have occurred where studs have been threaded for the full length, or where there is a reduced section between the threaded ends.

Tubes Subjected to External Pressures. The high-pressure converters have a pyrometer tube 2-in. OD × $^7/_8$-in. ID of Type 347 stainless steel. This tube carries the thermocouples for measuring the temperatures at various levels in the vessels. It was designed for external pressures of 10,000 psi and temperatures up to 900 F.

During a runaway reaction in one of these units, the temperature climbed rapidly, exceeding the 1500 F limit of the recorder. It was concluded from the grain size of the metal and the very small number of cracks, that the temperature had exceeded 1800 F while subjected to the 10,000 psi external pressure. This combination of pressure and temperature caused the tube to collapse, Fig. 19.

[7] "High-Pressure Injection Pumps in the Coal-Hydrogenation Demonstration Plant," by J. T. Donovan, B. H. Leonard, Jr., and J. A. Markovits, presented at the Annual Meeting, Atlantic City, N. J., November 26–30, 1951, of The American Society of Mechanical Engineers (Paper No. 51—A-71 unpublished).

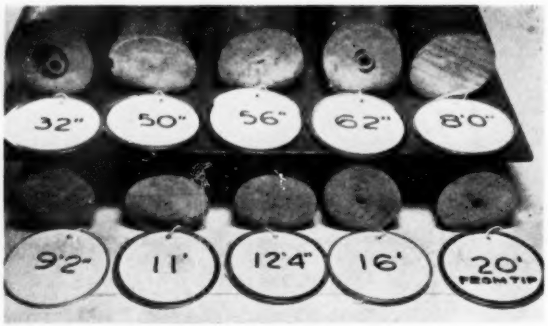


Fig. 19 Collapsed High-Pressure, High-Temperature Pyrometer Tube Showing Appearance at Different Distances From Tip

Closer temperature control through better cooling-gas distribution is expected to eliminate this hazard in future runs.

Conclusions

Although it is possible that other metallurgical problems may develop as plant operations continue, it is felt that the most serious difficulties have been discovered and solved satisfactorily. It is interesting to note that except for erosion at pressure-reduction points there was very limited metal deterioration in the high-pressure system. The corrosion in the low-pressure equipment, while appreciable, was no more than had been anticipated.

Acknowledgments

The authors wish to thank R. R. Hartnett, Mechanical Engineer, for his co-operation and able direction of plant-maintenance procedures during turn-around inspection. They also thank R. P. Meyerand, Chemical Engineer, for his collection and accurate compilation of supporting data.

Discussion

G. A. Nelson.[8] This paper has been read with a great deal of interest, and we wish to congratulate the authors on their ingenuity in correcting the troubles they have had with mechanical equipment operating under conditions of extremely high pressures and temperatures combined with erosive and corrosive environments. We have two specific comments to offer:

Under the heading, Hydrogen Attack, the limits for carbon and alloy steels are given and it is gratifying to find that after years of operation no trouble has been found with the piping when used under these prescribed limits. The authors show that when the limits for either carbon steel or 1 per cent chromium steels have been exceeded the effect from hydrogen is very rapid. This indicates how critically important is the proper amount of alloying elements for adequate resistance to hydrogen at high temperatures and pressures.

Under the heading, Stress Corrosion Cracking of Caustic Storage Tanks, we can confirm that the temperature limit for "as-welded" steel storage vessels handling caustic is very critical. We cite an example of a column in caustic service which had been handling dilute caustic at a temperature of about 190 F for 15 years without any trouble; however, operation of the column was found to be better when the temperature was raised to 240 F. Three months after the temperature was raised the column suffered caustic cracks adjacent to welds. It was removed from service and replaced by a stress-relieved vessel.

[8] Shell Development Company, Emeryville, Calif.

Authors' Closure

The interest which Mr. Nelson has shown in this paper is sincerely appreciated, particularly because of his familiarity with similar problems. His splendid assembly of data on hydrogen attack of carbon and alloy steels at high temperatures and pressures was referred to when the design limits for the steel in the hydrogenation demonstration plant were set. He subsequently suggested that lower alloy steels would be satisfactorily resistant for the intermediate temperature range. It was not known, however, if the mechanical properties of steels with lower chromium content would be satisfactorily stable when heat-treated to withstand the required fiber stresses. Babcock & Wilcox Tube Company has since investigated the effect of aging at 650 and 850 F on normalized and tempered Croloys $2^1/_4$, 5, and 7 (containing $2^1/_4$, 5, and 7 per cent chromium, respectively). In 5000-hr tests Croloy $2^1/_4$ was satisfactorily stable at 650 F and Croloy 7 was satisfactorily stable at 850 F. Preliminary exposure tests have confirmed the reported resistance of these steels at these same temperatures.

The case cited by Mr. Nelson of a column that developed caustic cracks adjacent to welds confirms the experience noted in the paper and further emphasizes the necessity for stress-relieving welded vessels operating at temperatures above 200 F in caustic service.

Pressure Operation of Large Pulverized-Coal-Fired Boilers on the American Gas and Electric System

By G. W. BICE[1] and W. M. YEKNIK[2]

In 1946 a detailed study was made of the possibility of applying a special pressure-tight steel casing to the boilers for two projected 150,000 kw capability units, for the purpose of permitting operation with forced draft only. Factors influencing the decision to proceed with this type of construction are reviewed in detail. Operating histories, as affected by the pressurized construction, are reviewed for these two units and for four subsequent additional units of similar design. The principal difficulties which were encountered are enumerated and measures which were taken to overcome these difficulties are discussed. Static-pressure operating levels, auxiliary power, and thermal efficiency comparisons are given for pressure operation versus suction operation.

Introduction

Early in 1946, plans were initiated by the American Gas and Electric Service Corporation for the design and installation of two 150,000 kw capability single-boiler compound turbine-generating units, one for the Twin Branch Plant of the Indiana & Michigan Electric Company and one for the proposed new Philip Sporn Plant to be owned jointly by The Ohio Power Company and Appalachian Electric Power Company.

In considering the design of boilers to be used, a study was made of the possibility of using a special pressure-tight steel casing which would permit operation with forced draft only. This type of operation, with its basic simplicity and associated economic advantages, had been well known for many years, and had been in commercial use for some time, principally on oil-fired marine boilers, but also more recently to a limited extent on small pulverized-coal-fired experimental boilers.[3]

Even in the early stages of the study, it was recognized that application of pressure casings to large pulverized-coal-fired stationary boilers and their auxiliaries would present many new and difficult problems. However, the possible long-range advantages were so attractive that it was decided to proceed with the pressurized construction for both of the new units.

As these units were expected to operate at an over-all average heat rate of 9270 Btu per net kwhr, which would be materially better than system average,[4] their incremental value would be such as to make outages of any nature very costly. Therefore it was concluded that conventional induced-draft equipment would be provided, but arranged for complete by-passing when it was desired to operate pressure.[5] Also, it was believed at the time that as pressure operation experience accumulated, design of future units could be undertaken safely without induced-draft equipment. While this is still a long-range objective, its attainment undoubtedly will be slower than at first projected, and may be postponed until boilers are designed for complete outdoor installation.

Advantages of Pressure Operation

The advantages associated with pressure operation are many. Those which accrue from the design of the boiler enclosure itself can be realized at all times, regardless of whether operation is pressure or suction. Many others can be realized only when pressure operation actually is being used. Still other advantages can be obtained only when and if induced-draft equipment is omitted entirely from the basic design; however, definite potential disadvantages are associated with this design concept and must be weighed along with the advantages in order to evaluate fairly such a possibility.

The more important advantages will be listed and explained briefly, according to the foregoing classification.

1 Advantages that can be realized under both pressure and suction operation:

Reduced setting leakage: The pressure casing effectively eliminates setting leakage, both outward (during pressure operation) and inward (during suction operation), except for a small controlled amount of cooling and sealing air. This permits operation with lower excess air than would be possible with conventional casing, and thereby makes possible an improvement in boiler efficiency of from 0.6 to 1.2 per cent for large pulverized-coal-fired central-station-type boilers. Approximately half of this efficiency improvement is due to a decrease in quantity of gas to the stack and half to reduced exit-gas temperature resulting from more effective heat transfer with the smaller total gas quantity passing over convection heating surface.

Improved superheat and reheat control characteristics: Gas mass flow over superheater and reheater convection tube banks will not change with time due to increased setting leakage, as is frequently the case with boilers having conventional casings.

Improved control of furnace conditions: The absence of variable setting leakage makes possible closer control of fuel-air proportioning in the combustion zone of the furnace.

Smaller flue-gas-outlet equipment: Capacity requirements of dust collectors, induced-draft fans (when installed), flues, and stack are reduced by an amount equal to the setting leakage which would be experienced with conventional casing, less the controlled cooling and sealing air.

2 Advantages that can be realized only under pressure operation:

[1] Steam Generation Section Head, American Gas and Electric Service Corporation, New York, N. Y. Mem. ASME.

[2] Manager, Twin Branch Generating Division, Indiana & Michigan Electric Company, Mishawaka, Ind. Mem. ASME.

[3] "Steamotive—A Complete Steam-Generation Unit, Its Development and Test," by E. G. Bailey, A. R. Smith, and P. S. Dickey, *Mechanical Engineering*, vol. 58, 1936, pp. 771–780.

[4] "Philip Sporn and Twin Branch Steam Electric Stations," by Philip Sporn, Trans. ASME, vol. 70, 1948, pp. 287–294.

Contributed by the Power Division and presented at the Fall Meeting, Chicago, Ill., September 8–11, 1952, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, July 3, 1952. Paper No. 52—F-32.

[5] The term "pressure" will be used throughout the remainder of this paper to denote operation with forced draft only.

Reduced auxiliary power: When operating pressure, all of the combustion air is handled at a relatively low temperature. When operating suction, the combustion air plus gas from the fuel is handled at a higher temperature, usually from 150 to 250 F above entering-air temperature.

Reduced auxiliary heat loss: When operating pressure, all of the heat loss due to draft-fan turbulence is added to the air ahead of the air heaters where approximately 40 per cent is recovered. When operating suction, approximately two thirds of the heat loss due to draft-fan turbulence is added to the flue gases leaving the air heaters where it is not recoverable.

Reduced fan maintenance: When induced-draft equipment is by-passed, fan wear is eliminated for all practical purposes. Even with relatively efficient dust-collecting equipment ahead of induced-draft fans, some wear is always experienced under suction operation.

Improved air-flow control: Automatic control is simplified by the by-passing of furnace-draft regulating equipment when operating pressure.

Safer operation during starting-up and low-load periods: Furnace fluctuations and the need for continual readjustment of fan-volume control to prevent hunting during starting-up and low-load operating periods is eliminated when forced-draft equipment only is used.

3 Advantages that can be realized only when unit is designed for pressure operation without stand-by induced-draft equipment:

Reduced boiler-plant investment: This results from elimination of material and installation labor for induced-draft equipment, with associated instruments and controls; and reduction in size of boiler-room building for indoor boiler units, or reduction in steel and foundation requirements for outdoor boiler units.

Basic simplification of automatic control system: This differs from the item, "improved air-flow control," in that no provision whatever need be made for furnace-draft regulation or forced-draft-fan trip interlocking.

Disadvantages of Pressure Operation

Were it possible to obtain perfect design and workmanship on the boiler and its auxiliaries and to assume faultless operation from the initial start-up throughout the life of the equipment, only one disadvantage could be assigned to pressure operation, this being associated with increased first cost. A number of other disadvantages which are theoretically avoidable, in all probability, will be encountered in actual practical design, workmanship, and normal day-to-day operation. Most of these can be minimized, however, if induced-draft equipment is installed, as it has been found that transfer from pressure to suction and vice versa can be accomplished in 2 to 3 min without difficulty, even under full-load conditions.

The actual disadvantages which are most likely to be encountered will be listed and explained briefly.

1 Unavoidable disadvantage:

Increased boiler-plant investment: This results from the incremental cost of pressure casing over conventional casing, seal and aspirating air piping, and special leakproof construction for ash hoppers, soot and slag blowers, dampers, automatic oil lighters, primary-air fans, and other auxiliary equipment.

2 Theoretically avoidable disadvantages:

Limited access: Only small doors (about 3 in. diam maximum) can be fitted with practical compressed-air aspirating facilities which permit opening under pressure operation.

Difficulty in locating tube leaks: Aspirating doors make so much noise that minor tube leaks cannot be heard through them.

Difficulty with soot and slag blowers: Required close-clearance air-sealed wall boxes are more likely to cause binding under conditions of slight misalignment or element warpage than are conventional wall boxes.

Difficulty with rotating-shaft seals: Development of completely reliable, trouble-free shaft seals progressed slowly at first. This problem was more difficult on large equipment, such as primary-air fans, flue-gas recirculating fans, and large louver dampers than it was on smaller equipment such as coal feeders and butterfly dampers. Improved shaft seals have been developed; it is believed future difficulties from this source will be few.

Difficulty with ash and slag removal: Visual observation of ash-hopper contents is limited, and access for manual breaking up of slag formations is restricted to small and sometimes inadequate aspirating-type doors. This difficulty is likely to be of greater concern on slag-tap units than on dry-ash-removal units, because of the design requirement that slag removal be accomplished without complete draining of ash-pit water.

Increased coal-bunker fire hazard: This disadvantage is of significance only when high-volatile coals, which are especially susceptible to spontaneous combustion, are used. When bunker fires are experienced, the higher air pressure acting on the bunkers through the furnace, burners, pulverizers, feeders, and bunker gates tends to accelerate the burning rate and thereby make it more difficult to control and extinguish fires. Furthermore, complete emptying of bunkers for thorough cleaning is difficult, and can be undertaken only under the most carefully controlled operating conditions.

Dust nuisance: Minor casing leaks, experienced during initial phases of operation and occasionally following shutdowns, create a house-cleaning problem because of the outward leakage of dust and fly ash. This disadvantage diminishes after the first few months of operation, as erection faults and points of inadequate expansion provision are located and corrected.

Pressure Boilers on AG&E System

Table 1 lists all pressure boilers in operation, under construction, or on order for the AG&E System as of June 1, 1952. All twelve of these boilers are arranged for both pressure and suction operation. All are housed completely except for the dust collectors on the 150,000-kw Philip Sporn and Tanners Creek units and the air heaters, dust collectors, and forced-draft fans on the more recent 200,000-kw units.

Operating Experience, Twin Branch Boiler No. 51

Initial operation of the 150,000-kw capability single-boiler compound turbine-generator unit at the Twin Branch Plant was started in July, 1949, with the unit being placed in commercial operation on August 22, 1949.

The steam-generating equipment consisted of a pulverized-coal-fired open-pass boiler with integral superheater, reheater, and economizer, two parallel regenerative air heaters, a multiple-cyclone-tube dust collector and conventional auxiliary equipment, including induced-draft fans but arranged with gas by-pass facilities to permit continuous pressure operation. The general arrangement of this equipment is shown in Fig. 1.

Originally it had been planned to operate this boiler under

TABLE 1 PRESSURE BOILERS ON AG&E SYSTEM

Plant	Boiler no.	Net capability of unit served, kw	Date of operation
Twin Branch	51	150000	July, 1949
Philip Sporn	11	150000	November, 1949
Philip Sporn	21	150000	June, 1950
Tanners Creek	1	150000	March, 1951
Philip Sporn	31	150000	August, 1951
Philip Sporn	41	150000	February, 1952
Tanners Creek	2	150000	Under construction
Kanawha River	1	200000	Under construction
Kanawha River	2	200000	Under construction
Muskingum River	1	200000	Under construction
Muskingum River	2	200000	Under construction
Tanners Creek	3	200000	On order

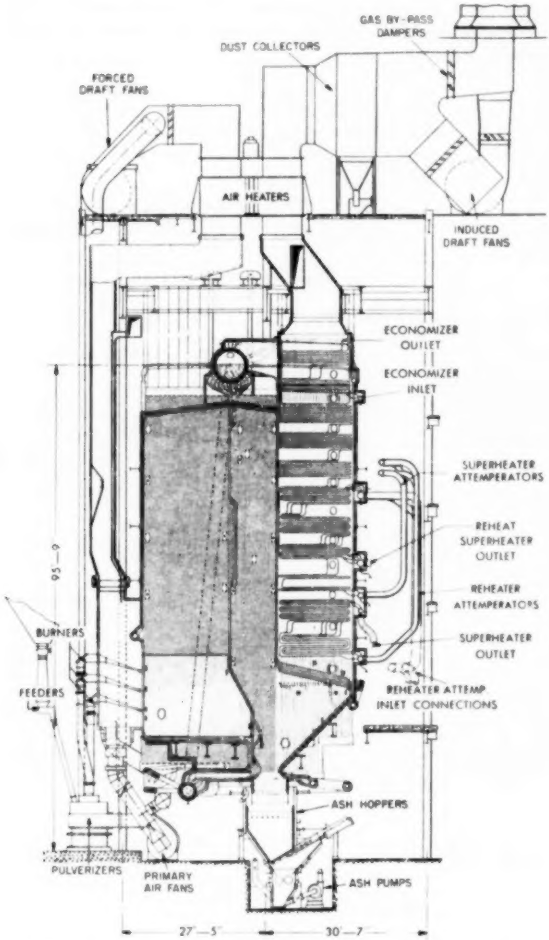


Fig. 1 Cross Section of Twin Branch Boiler No. 51

pressure from the initial start-up; however, unusually heavy capacity demands on the American Gas and Electric System necessitated an acceleration in the erection schedule. Because of this, thorough pressure testing of the setting prior to initial operation was impossible, and the boiler, as placed in service, was not fully prepared for pressure operation. In view of the heavy system load requirements and the consequent cancellation of a formal pressure-testing program, which would have required an additional 2 to 3 weeks, operation of the unit underwent three distinct phases in the conversion from suction to pressure operation:

1 The initial phase of operation was conducted almost exclusively under suction conditions using both forced-draft and induced-draft fans, in order to place the unit on the line at the earliest possible date. During this period, a few short trial runs were made under pressure to explore the general characteristics of this type of operation and also to determine the location of major setting leaks.

2 The second phase of operation consisted of a relatively long period with operation basically suction. During this time, however, frequent pressure runs were undertaken in order to study the various difficulties encountered under pressure operation conditions, to analyze the nature of all setting leaks, and to devise appropriate corrective measures.

3 The third phase, that of regular pressure operation, was begun approximately 2 years after the unit was first placed in service, and from that time until the present, operation of the boiler under pressure has approximated 95 per cent of the total availability time for the unit. The relatively infrequent periods of suction operation required during this time were principally for maintenance of auxiliary equipment.

During the initial operating period, work was continued to complete the inevitable odds and ends of erection which result from accelerated scheduling, and to correct obvious major leakage in the boiler setting proper. The latter consisted to a large extent of field work, principally in the nature of removing insulation, welding casing joints, and reinstalling insulation in the affected areas. Under conditions existing at the time, the only practical procedure to determine the degree of tightness of the setting was to place the boiler on pressure operation for short test periods and to check various areas of the setting by thorough visual inspection. Following each such test period, the unit was returned to suction operation and corrections were made to the faults noted. Due to the extremely large physical size of the unit, a considerable amount of time was consumed in this type of corrective procedure. During this initial operating period, other minor difficulties normally encountered on new installations also were corrected.

During the second operating phase, the unit was transferred to pressure for more extended periods of time and research was conducted to locate and analyze the difficulties being encountered under actual pressure operating conditions. Much time and effort were expended in studying these difficulties, in redesigning various setting-enclosure details, and in modifying appurtenances and auxiliaries, in order that all components of the equipment would conform to the rigorous requirements of pressure operation. Some of the factors encountered were redesign of expansion joints; modification of casing details; installation of supplemental seal air facilities; adaptation of slag-blowing equipment to pressure operation; installation of enclosures on dampers located in the fly-ash-laden gas stream; and many other similar details, all of which had to be given careful consideration in order to correct apparent weaknesses in the original design.

In the third phase, pressure operation of the boiler has been considered normal, the only exceptions being the infrequent short periods of suction operation necessary to accommodate maintenance requirements on certain auxiliary equipment, i.e., pulverizers, fans, and slag blowers. Redesign of this auxiliary equipment to permit the performance of routine maintenance work with the boiler operating pressure is now being expedited in order that this boiler may operate under pressure 100 per cent of the time. Present indications are that this will be accomplished in the near future.

The dates given in Table 2 highlight the transition from suction to pressure operation for this boiler.

Figs. 2 to 11, inclusive, cover some of the more important modifications and redesigns which were required to make pressure operation of this unit practical. For clarity, the illustrations have been arranged to show both the original design and the modified or final design.

Operating Experience, Philip Sporn Boilers Nos. 11, 21, 31, and 41

A cross section of boiler No. 21 at Philip Sporn Plant is shown in Fig. 12. Boilers Nos. 11, 31, and 41 duplicate boiler No. 21, except for the addition of two rows of tubes in the bottom bank of the primary superheater.

TABLE 2 TRANSITION FROM SUCTION TO PRESSURE OPERATION

Date	Operation	Condensed summary of operating history
7/12/49	Suction	Unit on line for 50 min; maximum load, 15,000 kw
7/12/49 to 7/20/49	Out of service	Inspection of steam-generating and turbine-generator equipment in preparation for normal commercial operation
7/20/49 to 8/21/49	Suction	Unit in intermittent operation, due to outages required for removing obstructions left in furnace wall tubes by boilermakers
8/22/49	Suction	Unit placed in commercial operation
9/24/49	Pressure	Unit on pressure operation with induced-draft fans out of service and by-passed for first time. Furnace pressure 15 in. of water. Operation continued on this basis for $4^1/_2$ hr, after which it was returned to suction operation because of excessive setting leakage
9/25/49 to 10/7/49	Suction normal; pressure for testing only	Intermittent suction and pressure operation to locate leaks
10/7/49	Pressure	Unit removed from service with boiler on pressure operation and induced-draft fans by-passed
10/9/49 to 2/13/50	Suction normal; pressure for testing only	Suction operation normal; boiler placed on pressure operation for longer periods to continue leakage testing and repair
2/14/50 to 4/15/50	Out of service	Unit out of service for 2 months for redesign of furnace-floor support structure, lowering of furnace outlet screen, removal of two loops of primary superheater tubes, and major redesign of casing on boiler and in furnace floor area; corrections also made to other parts of setting, such as expansion joints, access doors, seal air piping, damper shaft seals, etc.
4/16/50	Pressure and suction	First start-up under pressure with induced-draft fans by-passed. Returned to suction operation after 3 hr 40 min of pressure operation
4/17/50	Pressure	First complete 24-hr period of pressure operation
4/18/50 to 7/6/51	Suction normal; pressure for testing only	Suction operation again considered normal, but periods of pressure operation further extended so as to continue with detailed leakage testing and repair. During this time the boiler was taken out of service on several occasions (principally during low-load week-end periods) for major alteration work on casing and auxiliaries, in preparation for regular pressure operation
7/6/51 to 7/14/51	Pressure	Completed one full week of continuous pressure operation
7/14/51 to present	Pressure normal; suction for maintenance of certain auxiliaries only	During this period, pressure operation has been considered normal, with suction operation being required only during scheduled periods when certain auxiliary equipment must be cleared for maintenance work

Boiler No. 11 was first placed in operation in November, 1949. Because of the earlier unsuccessful pressure operation trials at Twin Branch, low-speed forced-draft-fan motors only were installed initially. In March, 1950, several short-duration partial pressure runs were made with forced-draft fans wide open and induced-draft fans throttled so as to produce a furnace pressure of approximately 11 in. of water at 120,000 kw net output. During these runs severe gas leakage and rapid overheating of retractable slag blower wall boxes was experienced.

During a short outage in September, 1950, high-speed forced-draft-fan motors were installed on boiler No. 11. Renewed attempts to operate under pressure were generally unsuccessful because of leakage through the convection-pass rear wall between the inlet screen and secondary superheater center cavity. An investigation revealed that the original tube and shaped-tile construction in this area had been damaged seriously by the action of slag blowers located approximately $2^1/_2$ ft from the affected area. During May, 1951, the lower rear wall was rebuilt using flat studs with poured-refractory backing in place of the original shaped-tile construction, the rear retractable slag blowers were relocated, and new improved wall-box seals were installed on all blowers. After several weeks of operation alternating between suction and pressure, during which time miscellaneous casing and door leaks were corrected, the boiler was placed on regular pressure operation.

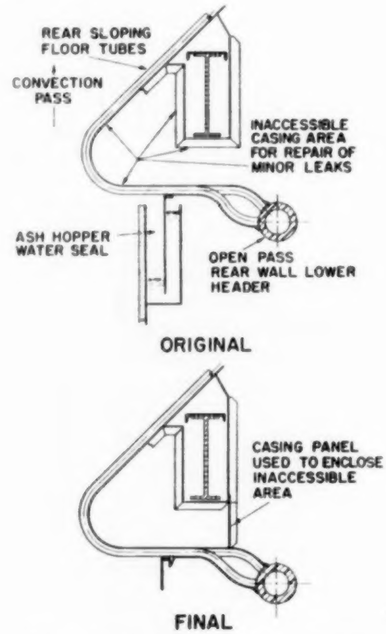


Fig. 2 Convection-Pass Floor Casing

(Trouble with casing failures in inaccessible areas behind sloping floor-support beam was eliminated by providing complete enclosure through installation of short external casing extension.)

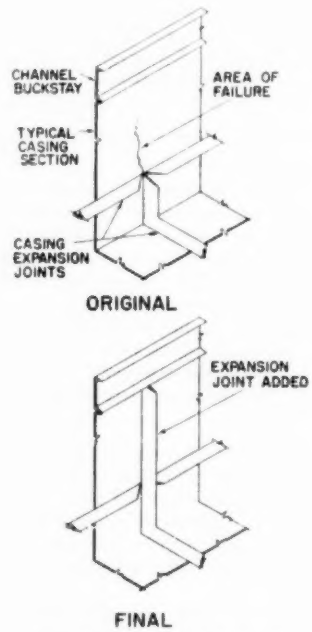


Fig. 3 Economizer Casing

(Lack of proper expansion facilities in casing panel was corrected through addition of single-bellows vertical expansion joint.)

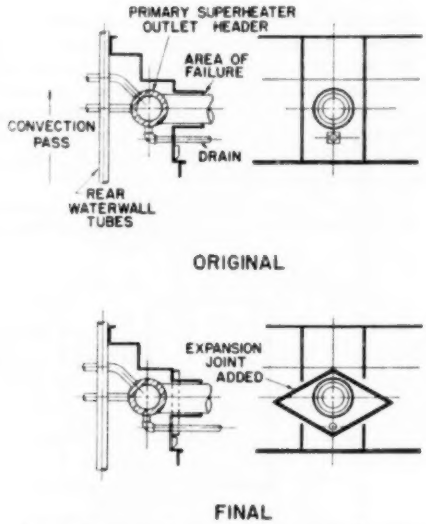


Fig. 4 Primary Superheater Outlet Header

(Expansion in transverse direction caused failure of casing at juncture with outlet pipe; correction was obtained by adding diamond-shaped single-bellows expansion joint.)

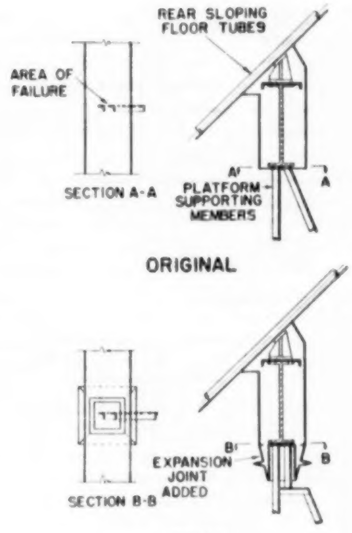


Fig. 5 Convection-Pass Floor Walkway Support

(Lack of expansion facilities for casing under convection-pass support beam at juncture with walkway support angles resulted in casing tears; installation of expansion joint eliminated failures.)

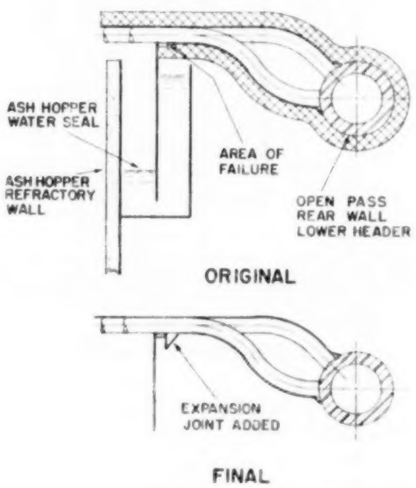


Fig. 6 Casing Seal at Ash-Hopper Water-Seal Skirt

(Differential expansion of generating tubes and casing caused failures in casing immediately outside of skirt; correction was obtained by addition of single-bellows expansion joint.)

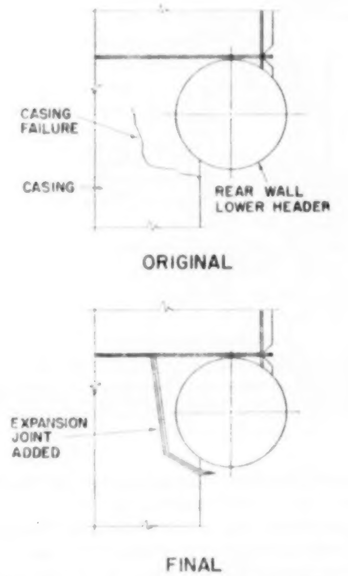


Fig. 7 Waterwall Header-Casing Tie

(Casing failure of tie between lower rear waterwall header and side-wall casing was corrected by adding expansion joint at approximate line of failure.)

Within a relatively short time, failure of the ash-hopper water-seal skirt made further pressure operation impossible and it was necessary to return to continuous suction operation. The seal skirt on this boiler was made in two parts, the submerged lower two thirds being composed of corragated stainless steel and the dry upper one third being made of plain carbon-steel plate. Investigation revealed that the carbon-steel portion had failed from extensive corrosion apparently caused by condensation of the sulphurous components of the gases from inside the ashpit. Since the corresponding water-seal skirts on Twin Branch boiler No. 51 had been single-piece stainless-steel corrugated plates and had given no difficulty, the Philip Sporn boiler No. 11 seal skirts were rebuilt using all stainless steel. Following this alteration, the boiler was again transferred to pressure operation.

As in the case of the Twin Branch boiler, pressure operation is now considered normal at all times except for short periods required for maintenance work on pulverizers, slag blowers, and for

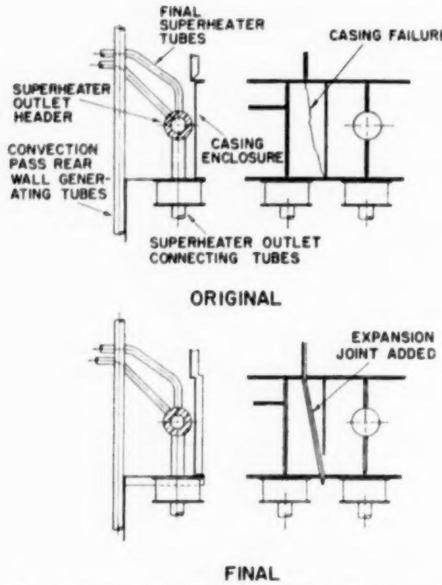


Fig. 8 Superheater Outlet Header Enclosure

(Casing failure in header enclosure was corrected by adding expansion joint at line of failure.)

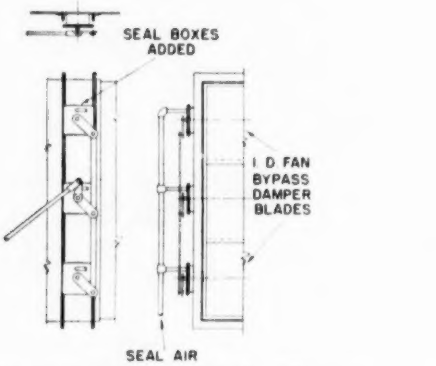


Fig. 10 Induced-Draft-Fan By-Pass Dampers

(Original construction, which is not shown, consisted of standard bearing enclosures which were not adequate to prevent leakage of gas and fly ash; correction consisted of installing individual air-sealed enclosures around each bearing.)

the occasional repair of minor leaks at casing expansion joints and around setting doors. It has been possible on a number of trials to perform maintenance work on pulverizers without changing the boiler to suction operation; however, special precautions have been necessary to seal connecting dampers. It is expected that this problem will be solved completely in the near future through the redesign of pulverizer-inlet dampers and outlet burner-line gates. In connection with slag blower removal, basic data on compressed-air removal equipment which has been completed in the laboratory has been applied to several blowers in the field recently, and it now appears simply a matter of time until all blowers can be equipped with practical compressed-air removal facilities.

One other difficulty, which had been experienced on this boiler (and on the other three Philip Sporn Plant boilers), involved removal of relatively large slag chunks which fell into the ashpits and could not be moved by the sluicing jets as originally installed. Manual lancing was often required to dislodge such slag chunks. Since lancing through small air-aspirating doors did not prove practical, it was found necessary to transfer the boiler to suction operation at unpredictable intervals and lance through the large access doors in order to dispose of this slag. The ashpit bottoms have been revised and the sluicing jets relocated recently in order to overcome this difficulty.

Boiler No. 21 at Philip Sporn Plant, which was placed in service in June, 1950, followed the same general pattern as boiler No. 11

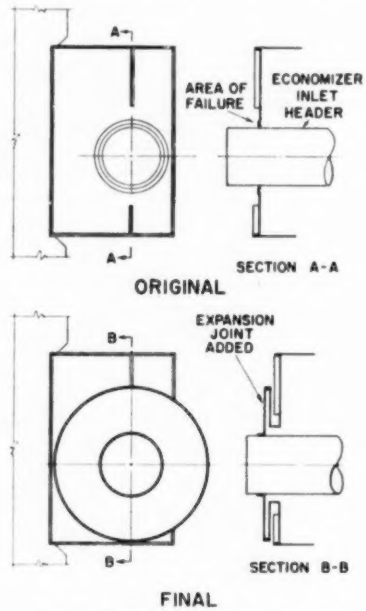


Fig. 9 Economizer Inlet Header

(Lack of adequate expansion facilities at juncture of economizer header and side-wall casing resulted in casing failure; installation of enlarged bellows-type expansion flange permitted full elongation of header without further failure of casing.)

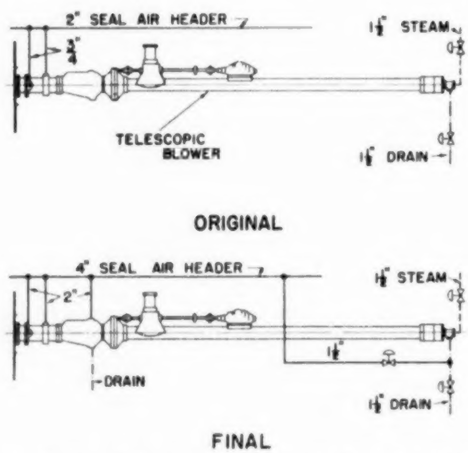


Fig. 11 Schematic Diagram of Telescopic Slag Blower Showing Modifications to Seal-Air Provisions

(Original installation provided insufficient seal air to prevent infiltration of fly ash into working parts of blower or to cool wall box effectively; correction involved modification of seal system to increase volume of air to wall box and also to supply air through steam passage of blower during idle periods.)

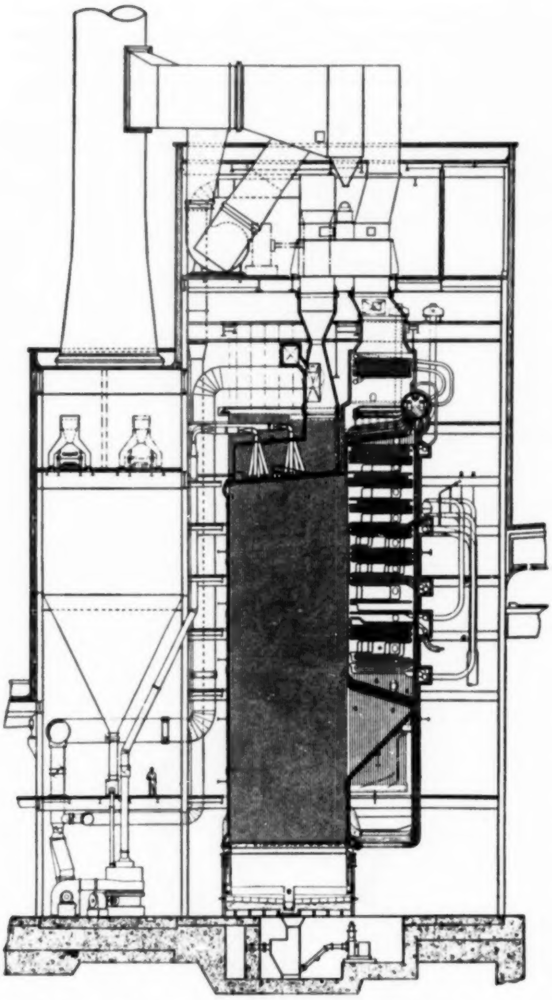


FIG. 12 CROSS SECTION OF PHILIP SPORN BOILER NO. 21

in its transition from suction to pressure operation. High-speed forced-draft-fan motors were installed; the convection-pass lower rear wall was rebuilt; slag blowers were relocated and equipped with new wall-box seals; the ash-hopper water-seal skirts were changed to all stainless steel, the bottoms reshaped, and the sluice liners relocated; and various casing expansion joints were replaced. Regular pressure operation was begun in October, 1951, and has been considered normal since that time.

In the case of both boilers Nos. 11 and 21, system-load requirements dictated an accelerated erection schedule which did not permit adequate pressure testing of the setting prior to initial operation. In the case of the next two Philip Sporn Plant boilers, Nos. 31 and 41, boiler erection was sufficiently ahead of turbine erection to permit from 1 to 2 weeks of setting pressure-testing prior to the boil-out.

The actual mechanics of testing required temporary blanking of air-inlet ducts, gas-outlet flues and all small openings such as fan shaft seals, damper shaft seals and slag blower wall boxes which eventually would be sealed by forced-draft air through an extensive system of permanent seal-air piping. After such closures were completed, internal pressure was built up to approximately 18 in. of water using a special connection from one of the pulverizer primary-air fans, temporarily equipped with a quick-closing blast gate. The time rate of pressure drop was then measured and, as this was found to be completely unsatisfactory initially, welded casing joints and other possible leakage sources were examined by coating with a test soap solution, after which pressure was dropped and all leaking joints rewelded. This procedure was repeated until a satisfactory pressure-drop rate was obtained. In the case of boilers Nos. 31 and 41, this was 5 in. of water pressure drop (from 18 in. to 13 in.) in 20 min. This rate of pressure drop was calculated to be equivalent to 140 cfm leakage, most of which probably occurred around temporarily sealed slag blower wall boxes, damper shafts, and other similar openings. Until the pressure-testing program was completed, no insulation was applied to casing joints or other areas of possible leakage.

To appreciate the tremendous effort required to obtain complete setting tightness, it must be realized that the total envelope subject to leakage on a boiler unit of this design consists of approximately 45,000 sq ft of external area. From this it should be apparent that very careful planning is required in order to establish a satisfactory erection and testing sequence for a pressure boiler unit.

Based upon experience on the previous units, combined with the thorough pressure testing made possible by the planned erection schedule, boiler No. 31 at Philip Sporn Plant was operated on pressure almost from the initial start-up. The only major difficulty resulted from a design change in furnace-inspection doors from the previously standard 3-in-diam circular doors to new 2-in. × 5-in. rectangular doors. It was quickly discovered that compressed-air aspiration through a rectangular opening was considerably less efficient than through a circular opening. With the unit at full load it was necessary to operate induced-draft fans and reduce furnace pressure from the normal 18 in. of water to approximately 10 in. of water in order to inspect the boiler through these rectangular doors. Correction of this difficulty is now in progress, through replacement of all rectangular doors with 3 in. circular doors, similar to those installed on boilers Nos. 11 and 21.

A delay in shipment of certain turbine parts for unit No. 4 at the Philip Sporn Plant permitted 10 extra days for pressure-testing boiler No. 41. Resulting tightness of the casing was so favorable that both the boiling-out and the setting of safety valves were accomplished without the use of induced-draft fans. This afforded an unusually good opportunity to examine the casing with the boiler hot, thereby permitting location and subsequent repair of leaks and faults produced by expansion, which otherwise could have been found only after the unit was in regular operation.

Both boilers Nos. 31 and 41 at Philip Sporn Plant (and also boiler No. 1 at Tanners Creek Plant) were installed with high-speed forced-draft-fan motors only. It has been found that starting-up can be accomplished satisfactorily with high-speed forced-draft-fan operation under either suction or pressure conditions. The initial doubt that high-speed fans could be throttled sufficiently by the use of the inlet vanes to permit the low-volume air flow required for starting-up was unfounded, as the fan manufacturer apparently has refined the design of inlet vanes on large fans so that volume control is effective down to 10 or 15 per cent of rated capacity. Thus, on all future pressure boilers, only high-speed forced-draft-fan motors will be provided.

OPERATING EXPERIENCE, TANNERS CREEK BOILER NO. 1

The first unit at the new Tanners Creek Plant of the Indiana & Michigan Electric Company, located at Lawrenceburg,

Ind., was placed in commercial operation in March, 1951. Initial plans for the boiler, which essentially duplicated the design of the Philip Sporn boilers, included provisions for future induced-draft-fan by-pass dampers, but with blanking plates to be installed initially. However, a few weeks before the boiler was placed in service, progress with pressure operation at Twin Branch was considered encouraging enough to warrant installation of the by-pass dampers. These were obtained on a rush basis and installed during the shutdown following the preliminary shakedown run for setting safety valves, balancing the machines and general checking, and tentative plans were made to begin pressure operation immediately thereafter. However, after the first few hours of operation, serious leaks developed in the casing in the large cavity below the two furnace center hopper slopes and it became necessary to return to suction operation for another several weeks.

Following a week-end shutdown during which the hopper slope casing leaks were repaired, the boiler was operated on suction during the day shift to facilitate welding of miscellaneous minor casing leaks, and on pressure during the remainder of each 24-hr period. After 2 months of operation with approximately 75 per cent of the time on pressure, difficulties developed at the lower section of the convection-pass rear wall, similar to those experienced on the Philip Sporn boilers, and pressure operation had to be abandoned temporarily.

Upon completion of rear-wall repairs, pressure operation was resumed, and shortly thereafter declared normal. The record of per cent of total time under pressure operation at Tanners Creek has been excellent. During the spring of 1952 this figure averaged about 95 per cent and during the early summer, increased to approximately 99 per cent.

Retractable Slag Blower Seals

One of the more interesting problems involved in developing practical pressure operation has been that of sealing full-retracting slag blower wall boxes.

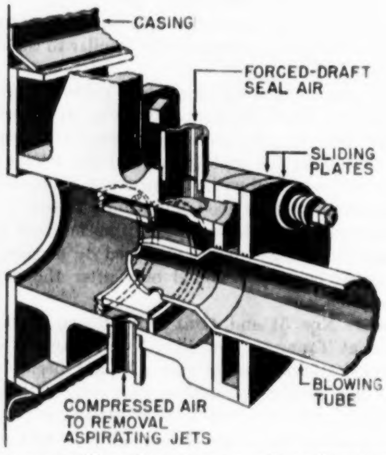


Fig. 13 Original Wall-Box Seal for Full-Retracting Slag Blower

The wall-box seal installed on the first full-retracting blowers, Fig. 13, was based upon sealing a close-clearance annulus, formed by the blowing tube and a concentric spring-loaded sliding plate, with forced-draft sealing air. This seal was satisfactory in some respects, but eventually had to be abandoned because of excessive clearance in the annulus, initial poor fit plus warping of the outer sealing plate, and pulling apart of the sealing plates against the springs because of blowing-tube fouling and misalignment. It also was apparent that the original seal-air piping was not large enough to supply the volume of forced-draft air required to prevent outward leakage of gas and fly ash.

In an attempt to improve the original seal as quickly as possible, a segmental cast-iron outer seal, held together by a small garter spring, was added. This seal, Fig. 14, proved unsatisfactory as the segments tended to be pulled out of place by the blowing tube on its return travel, and also wore both itself and the blowing tube excessively.

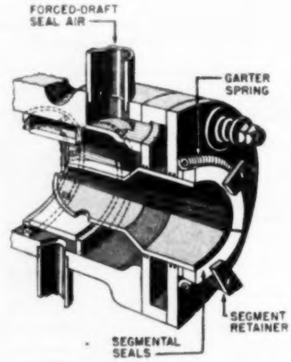


Fig. 14 Modification to Original Wall-Box Seal for Full-Retracting Slag Blower

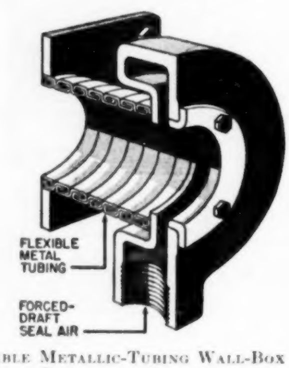


Fig. 15 Flexible Metallic-Tubing Wall-Box Seal for Full-Retracting Slag Blower

The next wall-box seal to be developed, Fig. 15, consisted of two parallel close-clearance annuli, connected rigidly by a sealing air chamber, and fastened to the wall box proper by a short length of flexible metallic tubing. This seal at first appeared to be entirely satisfactory; however, after a few months of operation it became apparent that it had several weaknesses. The seal-air chamber, which hung on the outer end of the flexible metallic tubing, had a tendency to sag and bind due to lack of support. Several guide and spring-suspension attachments were tried but none was entirely satisfactory. The tendency to bind also caused excessive wear of both the blowing tube and the air-seal annuli inner surfaces. As the annuli clearances increased from this wear, some outward flow of gas accompanied by sealing air led to local overheating of the brazed joint between the flexible metal tubing and the wall-box outer flange. In several cases this joint cracked and the seal failed completely.

At the same time that developmental work was proceeding on the flexible metallic-tubing seal, a completely different seal was

being designed. This seal, Fig. 16, employed multiple unrestrained close-clearance rings. The first of these seals which was installed at Tanners Creek gave satisfactory service for about 2 months. It was then removed and reinstalled on one of the

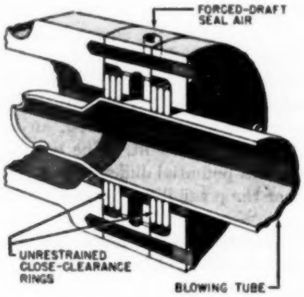


FIG. 16 MULTIPLE UNRESTRAINED-RING WALL-BOX SEAL FOR FULL-RETRACTING SLAG BLOWER

FIG. 17 RIGID COMBINATION WALL-BOX SEAL AND BLOWING-TUBE SUPPORT FOR FULL-RETRACTING SLAG BLOWER

Philip Sporn boilers along with four more seals of similar design. It soon developed, however, that these seals were not operating as designed owing to the fouling of the unrestrained rings with fly ash. Maximum life of these seals was about 3 months, with severe gas leakage and blowing-tube wear occurring during the last month.

Two additional seals, Figs. 17 and 18, are now in the developmental stage, with an experimental model of the first having been installed for about 2 months on one blower on a trial basis. Both of these seals have the dual function of supporting the boiler end of the blowing tube and of sealing against outward gas leakage. By providing for support of the blowing tube in the seal itself rather than employing a separately mounted external bearing, excessive relative movement between the blowing tube and the seal has been eliminated. Present indications are that the seal shown in Fig. 18 should provide the most satisfactory solution to this perplexing problem.

FUEL-HANDLING PROBLEMS

The handling of coal from bunkers to burners on pressure boilers has presented several unusual problems, some of which still are not solved completely. To appreciate fully these problems, it is necessary to consider the air pressures obtaining throughout a typical pulverized-coal system under both pressure and suction operating conditions. These are listed in Table 3 for a boiler of the Philip Sporn design.

TABLE 3 FULL-LOAD AIR PRESSURES, PHILIP SPORN DESIGN BOILER

Location	Suction operation air pressures, in. of water	Pressure operation air pressures, in. of water
Primary-air fan inlet	+ 1	+20
Primary-air fan outlet	+26	+45
Pulverizer upper housing	+12	+31
Burner-line shutoff gate	+ 8	+27
Burner windbox	+ 2	+21
Furnace	− 0.5	+18.5
Raw-coal feeder, pulverizer in operation	+11	+30
Raw-coal feeder, pulverizer off	+ 2[a]	+21[a]

[a] These pressures assume primary-air shutoff dampers open; with dampers closed, pressures will approach zero, depending upon relative tightness of dampers.

In order to open a pulverizer for inspection or maintenance work it is necessary to obtain a reasonably tight shutoff on the primary-air supply and a practically watertight shutoff on the burner lines. On all six of the subject units, special close-clearance butterfly dampers were supplied initially for primary-air shutoff. Since these dampers were subject to temperature variations of approximately 500 F, binding was frequently experienced; consequently, it was necessary to increase clearances by burning or grinding the edges of the disks. In most cases, the resulting free fit permitted excessive primary-air leakage. To correct this, a number of new damper designs have been developed. Several of these are under field test, and it now appears that at least two will be satisfactory.

Rubber-seated side-hinged swing gates were provided originally in the burner pipes to obtain the required tight closure between

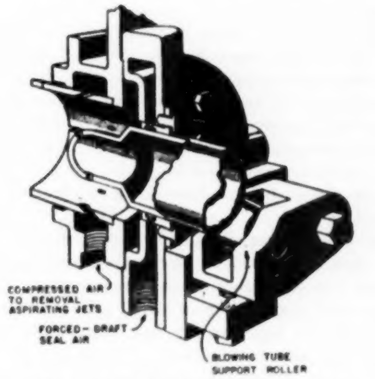


FIG. 18 RIGID WALL-BOX SEAL WITH ATTACHED BLOWING-TUBE SUPPORT ROLLERS FOR FULL-RETRACTING SLAG BLOWER

pulverizers and the furnace. It soon became apparent that many of these gates did not close tightly enough to prevent the backward flow of furnace gases during periods when the corresponding pulverizer was shut down and the access doors opened for maintenance or inspection. Investigation revealed that the profile of the seat was generally unsatisfactory for dependable tight closure and also the 50 Durameter rubber seating material was too hard. Corrections involved redesign of the seat profile and substitution of 25 Durameter rubber to provide increased seat elasticity.

The removal of obstructions such as boards, gunny sacks, and other foreign material from raw-coal feeders is a disagreeable task under either suction or pressure operation. It is obviously worse under pressure operation because of the higher air pressures involved, and can be accomplished with safety only by providing the operator with a glass-visored hood. This problem is being solved by fitting feeder-outlet spouts with recessed shutoff gates, so as to permit rapid isolation of a feeder from its pulverizer with-

out the necessity of shutting down either the pulverizer or its primary-air fan.

The pulverizers on all of the subject units have been located at basement-floor elevation and the boilers suspended from structural steel approximately 100 ft higher. Thus it has been necessary to provide for differential expansion in the burner pipes, this being 3.8 in. for the Twin Branch boiler and approximately 1.3 in. for the Philip Sporn and Tanners Creek boilers.

The relatively low elevation of burners on the Twin Branch boiler resulted in the unfavorable combination of short burner pipes plus high differential expansion. To accommodate this, a large corrugated-rubber expansion joint was provided initially in each of the burner pipes. Shortly after the initial start-up, these expansion joints became a source of grave concern because of their susceptibility to destruction by fire. Since the joints were on the furnace side of the burner-line shutoff gates, failure of any one joint would release furnace gases under full furnace pressure and, until burner-line shutoff gates were closed, pulverized coal under full pulverizer pressure, directly into the boiler room. Such a failure might ignite adjacent expansion joints and result in very serious fire damage to adjacent equipment. To eliminate this potential hazard, a set of expansion joints containing a minimum amount of rubber has been installed on one pipe and a new joint composed entirely of heavy-duty flexible metallic tubing has been installed on an adjacent pipe. Both of these arrangements have been found to be basically sound, and a study is now being conducted to determine which actually can be applied to the remaining burner pipes under the existing limited space conditions.

The relatively small burner-line expansion on the Philip Sporn and Tanners Creek boilers is taken through four conventional rubber-insert couplings per burner. Excessive leakage of these joints has occurred from time to time, presumably due to the unusually high burner-line pressures. This has been corrected by covering the original rubber-insert couplings with light-gage U-shaped steel expansion joints welded directly to the burner piping. On future boilers, however, a single joint of improved design will be used.

Sealing of automatic oil lighters was expected to be a difficult problem when first considered during the design stages; however, the original 2-in. sealing-air connection per lighter combined with a stainless-steel gravity-closing shield gate between the burner windbox and the atomizer tip has proved adequate. Actual difficulties with automatic oil lighters have been typical of those encountered on conventional suction boilers, and include binding of moving parts in their guides, short circuiting of high-voltage current in the ignition-spark circuit, and fouling of sprayer plates by dirt in the oil. Removal of atomizers is possible with the boiler operating under pressure, but is accompanied by the discharge of high-pressure sealing air plus accumulated fly ash, which makes the task somewhat unpleasant. On future pressure boilers, compressed-air aspirating rings will be installed in the lighters to eliminate this objectionable feature.

One of the most serious potential problems in connection with pressure operation of pulverized-coal-fired boilers involves fires in the raw-coal bunkers. Actually, such difficulties have been experienced only on Twin Branch boiler No. 51, and these infrequently. Bunker fires, which result either from the loading of burning coal from storage or from spontaneous combustion of coal in dead spaces within the bunker tend to be intensified by the constant seepage of warm high-pressure air from the pulverizer upper housing through the feeders into the bunkers. As ignited coal reaches the restricted area at the bunker outlet, intensity of burning increases rapidly, and in extreme cases overheating of the bunker-outlet gate may result. This problem has been studied carefully and several tentative solutions proposed. These currently are being analyzed for possible application to the Twin Branch boiler. Fortunately, this difficulty appears to be restricted to the lower grades of high-volatile coals which are especially susceptible to spontaneous combustion.

Another problem which could be serious on boilers not equipped with induced-draft fans is that of pulverized-coal-burner coking. When and if coking occurred, the burner would have to be taken out of service and left out until a boiler outage could be arranged, provided the coked condition had been discovered before serious overheating had taken place. If a burn-through of any part external to the burner windbox occurred, an immediate forced outage of the boiler would result. The only practical method known to combat this potential difficulty is to design the burning equipment so that the possibility of coking in operation is minimized.

This problem might disappear altogether, however, were the boiler to be designed with cyclone burners[6,7] in place of conventional pulverized-coal burners.

Effect on Heat Rate

The effect of pressure operation on unit heat rate will vary with different steam-generating units. A greater improvement will be obtained on a boiler designed for relatively high draft loss than on a boiler designed for low draft loss. For example, assuming two boilers, each serving a turbine of 160,000-kw gross capability, the air volume to the forced-draft fans would be 320,000 cfm at 130 F, and the gas volume to the stack 460,000 cfm at 330 F, based upon a heat rate of 9200 Btu per net kwhr. Assuming two different sets of air resistance and draft-loss characteristics, fan power required would be as indicated in Table 4.

TABLE 4 FAN-POWER REQUIREMENTS

	High draft-loss boiler	Low draft-loss boiler
Air resistance, in. of water	11	7.5
Draft loss:		
Pressure operation, in. of water	15	9
Suction operation, in. of water	17[a]	10.5[a]
Forced-draft fan power:[b]		
Suction operation, kw	725	495
Induced-draft fan power:[b]		
Suction operation, kw	1610	995
Total fan power, kw	2335	1490
Forced-draft fan power:[b]		
Pressure operation, kw	1710	1090
Fan-power reduction, kw	625	400
Heat-rate improvement, Btu/net kwhr	38	24

[a] Tests have shown draft loss to and from induced-draft fans to be as high as 2 in. of water, due to flue turns, dampers, and fan inlet losses.
[b] At 60 per cent fan efficiency and 95 per cent motor efficiency.

These fan power savings do not represent total improvement which would be obtained in over-all thermal economy resulting from pressure operation, as the increase in sensible heat made available to the air heaters through the higher power from the forced-draft fans during pressure operation is not included.

Actual comparisons of performance under conditions of pressure and suction operation have been obtained at the Philip Sporn, Tanners Creek and Twin Branch Plants. A comparison of Tanners Creek boiler No. 1 under pressure operation with Philip Sporn boiler No. 11 under suction operation indicated an improvement in unit heat rate of approximately 55 Btu per net kwhr. A similar comparison on Twin Branch boiler No. 51, under pressure operation and suction operation, showed an improvement in unit heat rate of 93 Btu per net kwhr.

The former heat rate comparison represents that which would

[6] "The Horizontal Cyclone Burner," by A. E. Grunnert, L. Skog, and L. S. Wilcoxson, Trans. ASME, vol. 69, 1947, pp. 613–634.
[7] "Cyclone-Fired Pressurized Steam Generator," by Merle Newkirk, Trans. ASME, vol. 73, 1951, pp. 215–224

be used in making preliminary studies for the purpose of choosing between pressure and suction operation, as the Philip Sporn boiler was equipped with 875-rpm forced-draft fans designed only for suction operation at the time of the test, while the Tanners Creek boiler was equipped with 1170-rpm forced-draft fans designed for pressure operation. In the second comparison, the Twin Branch boiler was equipped with 1170-rpm forced-draft fans during both tests, and the auxiliary power required for suction operation was high due to these fans operating at partial load with a consequent reduction in fan efficiency.

Conclusions

Based upon the relatively short-time operating experience gained on the six pressure units in service to date, the following general conclusions are offered, with the understanding that they are tentative and may require revision in the future as additional operating experience accumulates:

1 Pressure operation of large pulverized-coal-fired boilers is now entirely practical, provided several short-duration shutdowns for correction of minor leaks and other difficulties can be tolerated during the first few months of operation.

2 Air-pressure testing of the entire setting of a pressure-boiler unit prior to initial start-up is essential, in order to prevent serious gas and fly-ash leakage which will necessitate frequent pressure-suction transfers or, in the case of boilers without induced-draft equipment, outages for corrective sealing work.

3 On outdoor boilers, the gas and fly-ash leakage problem, which is likely to be troublesome during early stages of operation on indoor boilers, should become almost insignificant. While this would appear to be a definite advantage for outdoor pressure units, it might conceivably result in "hot" furnace leaks being ignored until serious damage to casing or insulation had occurred, and thus lead to unscheduled outages; however, this could and should be avoided by vigilance on the part of operating and maintenance personnel.

4 The only significant difficulties remaining at the time of this writing involve pulverizer maintenance, raw-coal feeder cleaning, slag-blower sealing, and bunker fire protection. Practical solutions for the first three of these difficulties are in process and should be realized in the near future. A completely satisfactory method of protecting against bunker fires seems to be somewhat more distant at present, although fortunately, the problem appears to be serious only in the case of those particular high-volatile coals which are unusually susceptible to spontaneous combustion.

5 As soon as satisfactory solutions are obtained for the remaining difficulties (item 4) and these and other recent developments have proved their soundness over a reasonable period of time, omission of induced-draft fans probably will be warranted on future boilers that are not expected to be subject to heavy slagging or ash removal difficulties.

6 When heavy slagging or ash removal difficulties are anticipated, because of projected use of unusually low-fusion coals or for any other reasons, boilers should be equipped with stand-by induced-draft equipment in order to decrease the number of unscheduled boiler outages and thus increase unit availability.

7 On large reheat units, an improvement in unit heat rate of from 30 to 60 Btu per net kwhr should be realizable from draft-fan power reduction and heat-loss recovery through the use of pressure operation as compared to suction operation. A further improvement of from 50 to 100 Btu per net kwhr also should be realizable through the use of pressure-tight casing with its controlled sealing-air intake as compared to a conventional boiler casing with its uncontrolled setting air infiltration.

Discussion

M. H. Kuhner.[8] The information made available with this excellent paper is especially welcome at this time when many prospective boiler purchasers are seriously considering commitments for boilers to be constructed for pressure operation. The advantages of this method of operation are not questioned for purely gas-and-oil-fired installations and many such units are now under construction. That pulverized-coal-firing would present a number of new problems and that not all of them could be foreseen by the designers is understandable. None of the difficulties reported in the paper is unsolvable. The experience with these pulverized-coal-fired pressure-operation boilers should serve to encourage further work along this line.

In learning of the operating and maintenance problems reported the obvious thought occurs that a large part of the difficulties experienced at Twin Branch and Philip Sporn Plants can be prevented, or at least greatly reduced with new installations by:

1 More liberal proportion of air and gas passages. This means higher initial cost of the boiler, but saving in fan power alone should amortize the added investment. Lower resistance in flow of air and gas results in lower pressure in ducts and setting and thus a simpler sealing problem.

2 Dry-bottom furnace and furnace cooling sufficient to permit slag-free operation with coal of lowest ash-softening temperature. The first cost of the dry furnace may be slightly higher than that of a slag-tap furnace, but the saving in eliminating costly ash hoppers, ash pumps, elaborate seals, and cost of maintenance of this equipment will more than pay for the larger furnace. Integral pressure-type dry-ash hoppers of the type shown in Fig. 19, herewith, can then be employed. It is of course recognized that the final disposal of dry, powdery ash produced with dry-bottom furnace operation is more difficult than that of the granular ash from slag-tap furnace.

3 Coal feeders of a design which will produce a positive pressure seal between coal bunker and pulverizer (certain types of screw feeders will fill the bill), tight shutoff valves in coal pipes between pulverizers and burners and in primary-air pipes ahead of pulverizers. One such pulverizer installation was placed in service a few months ago supplying coal to a drying furnace under 45-in. wg positive pressure. A suitable coal feeder and positive shutoff valves in coal and primary-air pipes prevent coal and air leaks through the idle mill.

Benefits derived from a completely tight boiler setting should rightfully not be credited to pressure operation. Such benefits (virtual complete absence of setting leakage, permanent maintenance of design efficiency and capacity, lower fan power, lower setting maintenance expense) are being realized with all boilers equipped with correctly designed steel casings and many such were installed long before the thought occurred to operate large-capacity boilers with plus pressure in furnace and setting. The all-welded pressure-tight casing skin is a must for pressure operation, but such construction will pay for itself as well with boilers designed for suction operation.

The intent of Table 3 of the paper may be misunderstood. The authors probably wish to show that the saving in fan power by pressure operation as compared with suction operation is greater for a high-draft-loss boiler than for a low-draft-loss boiler. It is questioned that the intention is to suggest that a better over-all station heat rate is obtained with a high-draft-loss boiler than a low-draft-loss boiler. Higher auxiliary power

[8] Vice-President-Engineering, Riley Stoker Corporation, Worcester, Mass. Mem. ASME.

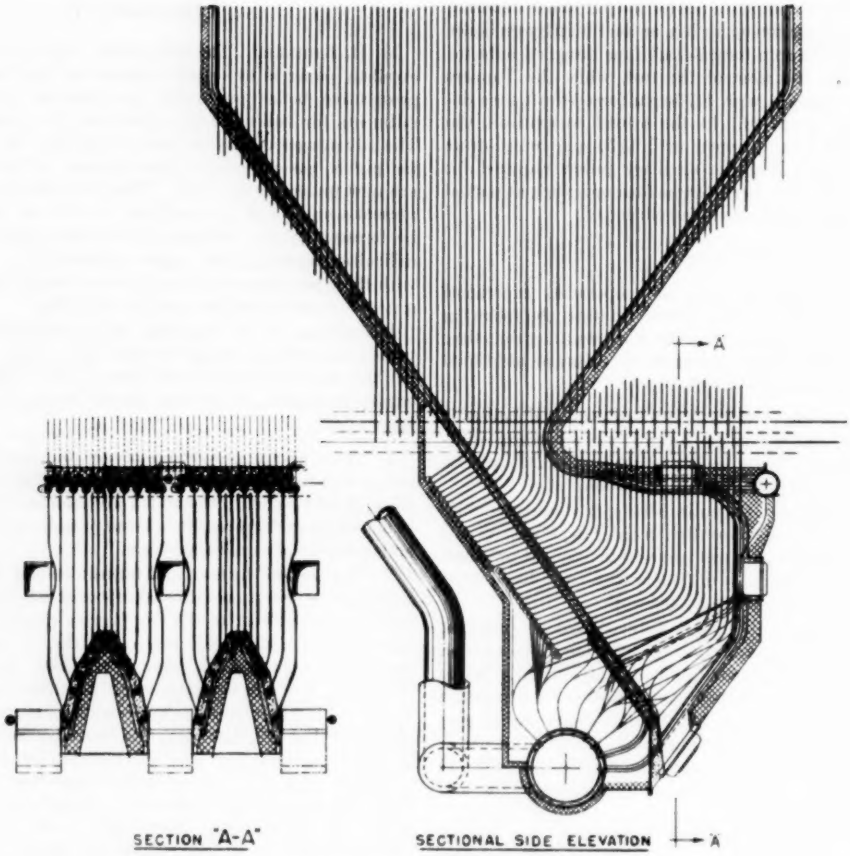


Fig. 19 Integral Ash Hopper

consumption for the high-draft-loss boiler would indicate that the opposite is the case.

P. R. Loughin.[9] We believe the authors have ably described the advantages and disadvantages of pressure-firing based on the experience of the American Gas & Electric system. The information on the percentage of time on pressure operation of the units shows a marked trend toward improvement of both design and workmanship on the later units. In addition, the authors emphasize the very considerable importance of a proper cold air-pressure test prior to actual operation of the unit and indicate such testing would require an additional period during erection of 2 to 3 weeks. This is true for a 1,000,000-lb-per-hr unit being erected on single-shift work 5 days a week. We suggest a better way of stating this requirement would be 10 to 15 shifts of 8 hr for units over 500,000-lb-per-hr capacity and somewhat less on smaller units.

We believe that Fig. 20, herewith, will be of interest to the readers of this paper since it shows graphically the per cent of operating time on pressure operation of the American Gas & Electric units and other successful units on pressure operation. The upper graph shows oil-and-gas-fired units and the lower graph coal-fired units. The first oil-and-gas-fired unit was placed in service in January, 1950, and since then six additional units have been successfully placed in regular commercial operation. None of these units is equipped with stand-by induced-draft fans and no forced outages have occurred because of difficulty with pressure casing or pressure operation. In addition to the six American Gas & Electric pulverized-coal-fired units referred to by the authors, the lower chart shows four cyclone-furnace-fired units, all of which are smaller units. These units are not equipped with stand-by induced-draft fans and they have all been in successful commercial operation, the first one starting up in February, 1949.

The authors bring out the fact that to date the access for hand-cleaning of any pressure-fired boiler is very limited since the only practical aspirated door for furnaces operating at relatively high pressures has been a 3-in. round door. The Babcock & Wilcox Company has expended considerable research effort to develop a wide-angle lance door for pressure units. The door shown in Fig. 21 of this discussion is the result of this research. This door has been tested in the laboratory and will successfully aspirate against very high furnace pressures. It will shortly be ready for field-testing.

W. L. Wingert.[10] This paper presents a valuable story

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[10] Mechanical Engineer, Mechanical Engineering Division, The Detroit Edison Company, Detroit, Mich. Mem. ASME.

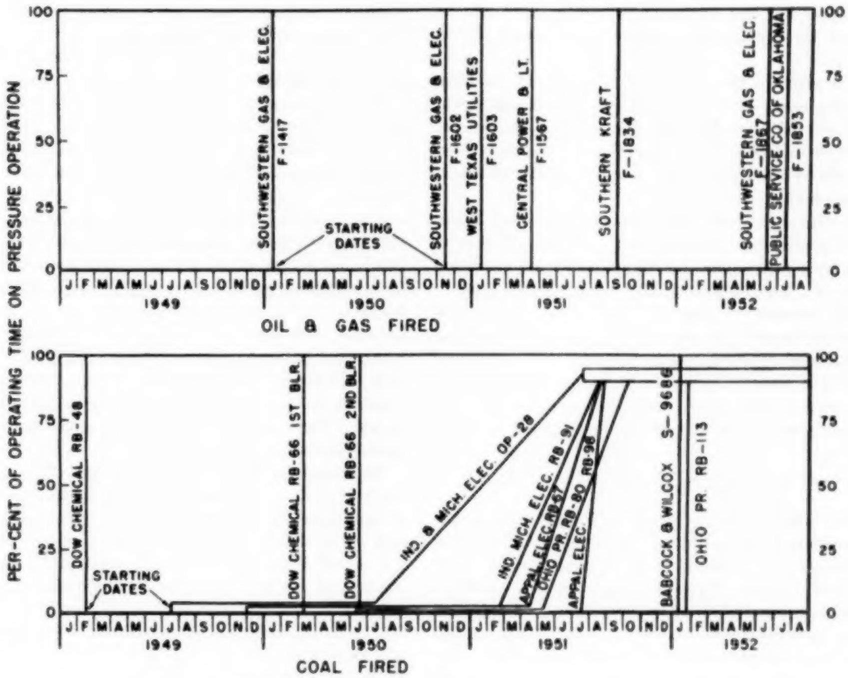

FIG. 20 PRESSURE-FIRED BOILERS IN SERVICE

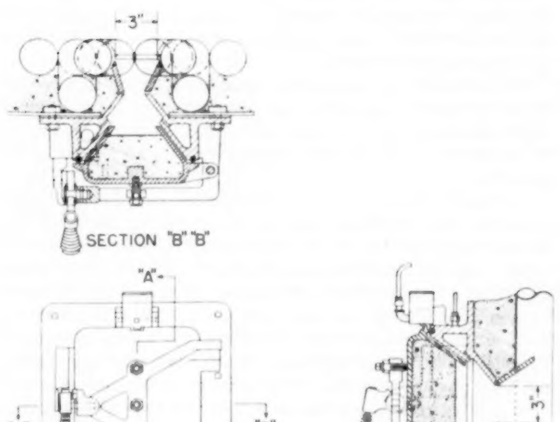

FIG. 21 WIDE-ANGLE PRESSURIZED LANCE DOOR

for many of us who are interested in lowering power-generation costs and who have not yet ventured into the field of pressurized furnaces. Of even more value to us, would be a more detailed discussion of the economics of this type of firing, covering differences in investment costs, in maintenance costs, and in availability.

The authors point out that reduced setting leakage can be realized when operating either pressure or suction. To provide for pressure operation, considerable money must be spent for improved casing design, expansion joints, ash gates, including a gate-cooling system, and soot-blower seals. An expensive seal air-piping system and additional compressor capacity are required. It seems that a major portion of the net saving available may be obtained by close attention to the design of a conventional casing and by frequent careful leakage checks by the operating personnel.

Modern units, owing to capacity needs and incremental value, generally speaking, operate at full capacity almost continuously. This type of operation, coupled with many operating variables such as fuel quality, feed, and preparation, occasionally will result in slagging, furnace-bottom-ash removal difficulties, coked burners, and so on. These problems will result in boiler outages if no induced-draft fan is provided. It seems likely that the cost of these outages will more than offset the saving made by omitting the fan.

The problems associated with the coal-feeding system are not easily solved. Tight shutoff dampers in air supply to mills are essential on certain types of mills feeding vacuum furnaces. Mill and burner-line fire problems are aggravated by hot-air leakage during mill outages. This becomes more critical with increased air pressure. Pressure furnaces, in addition to more expensive air dampers, require improved coal shutoff dampers, burner-line expansion joints, and burner construction.

Maintenance costs must be considered when justifying pressure operation. Fuel-system repairs will cost more as a result of tighter damper requirements and more severe service as pointed out earlier. Added compressor requirements and complicated seal air-piping system and seals will add to the plant labor bill. More diligent attention to casing maintenance is necessary. Certainly dust leakage will not be controlled perfectly at all

times; thus housekeeping and motor-maintenance costs will be higher. Offsetting these increases is reduced induced-draft-fan maintenance.

Many of us lose sight of the intangible savings realized from providing operating ease. Without question, the hazards and hardships placed on the operating crew as a result of pressure operation will result in less careful inspection of operating conditions. This leads to lower efficiency, more frequent and longer outages, and higher maintenance costs.

Summarizing, after increased maintenance costs and losses due to lower availability are charged, the net saving resulting from lower plant heat rate should be compared to the increased investment charges. The increased investments include more expensive casings, expansion joints, observation doors, ash gates, soot-blower seals, seal air piping, mill air- and coal-piping installation, and added air-compressor capacity. This increase will be reduced by a smaller induced-draft fan, or elimination of this fan. It may well be, that when all factors have been evaluated, little or no saving will be realized from pressure operation.

Authors' Closure

In Mr. Loughin's discussion, the graphs of Fig. 20 are interesting as they present a realistic picture of the progressively shorter times required to achieve practical pressure operation on large pulverized-coal-fired boilers. Pressure operation on gas-fired boilers, and probably some oil-fired boilers, should be achieved more easily than on pulverized-coal-fired boilers because of the absence of pulverizing equipment and either the complete absence of slag blowers or the utilization of nonretracting blowers on which wallbox sealing should be relatively simple.

The inspection door shown in Fig. 21 should improve the inspectability of the furnace and the convection-pass inlet area. It is hoped that this door will be perfected and field-tested soon, so that it will be available for installation on pressure boilers in the near future.

Two of the points raised by Mr. Kuhner involve basic boiler-design problems of a long-standing controversial nature, i.e., the balance between fan-power savings and the required higher investment cost of additional heating surface for low-draft-loss boilers; and the economics of a large conservative dry-bottom furnace versus a smaller less expensive slag-tap furnace for use with low-ash-fusion coal. As the effect of pressure operation on these problems is of a secondary nature, any discussion regarding the pros and cons of these questions would be beyond the intended scope of the paper.

When considering low-draft-loss boilers, it is obvious that a reduction of internal setting pressure should simplify the problems associated with sealing air and more important (although not mentioned in the discussion) aspirating air. This is not, however, of sufficient practical value to warrant any sizable additional investment, as these problems have now been solved satisfactorily for furnace pressures up to about 22 in. of water.

Referring to the integral ash hopper shown in Fig. 19 of the discussion, it would be interesting to speculate on just how ash might be removed from such a hopper through the several large "dry" doors, with the furnace under a pressure of 20, or even 10, in. of water. Also, it is doubtful that such doors could be made completely gastight and still be operable day after day.

As was stated in the paper, dampers and gates suitable for isolating coal pulverizers were being developed; it might be added that experience gained since the paper was presented has confirmed the prediction that such equipment could be altered or redesigned so as to give satisfactory operation.

To dispel the idea that present-day designs of conventional suction settings are tight, it is suggested that any one who actually believes this should try operating one of their suction boilers for a one-hour period at partial pressure, by reducing rating (if necessary), opening forced-draft fan controls wide, and throttling induced-draft fan controls to maintain proper steam flow-air flow ratio, until furnace pressure is several inches of water on the plus side. It is predicted that leaks will be found at many locations where it was not believed that leakage could possibly take place, and that the answer will become obvious long before the one-hour period is up.

In regard to Table 3, Mr. Kuhner has interpreted the table correctly; i.e., "The savings in fan power by pressure operation as compared with suction operation is greater for a high-draft-loss boiler than for a low-draft-loss boiler." It would seem that any other interpretation could result only from an incomplete reading of the paper, or possibly a lack of understanding of the subject.

The pressure-operation experience upon which Mr. Wingert's statements are based is not known. However, as applied to the boilers described in the paper, some of the statements are incorrect.

No ash-gate cooling system is required, as these gates are completely submerged, except during ash-removal periods on the dry-bottom boilers; and no special cooling arrangement is required during such periods.

No additional air-compressor capacity should be required because of pressure operation and none has been installed nor is contemplated on the new units, over and above that which would be furnished for a conventional suction boiler unit of comparable capacity.

In regard to increased boiler maintenance costs due to pressure operation, the predictions made in the discussion should be considered applicable only to the first year or so on the initial pioneer units. Actually, as experience is being obtained and cost figures analyzed, it appears that maintenance on pressure boilers may be actually lower in the long run, owing to the relatively infrequent use of induced-draft fans and to the improved design of various auxiliaries (necessitated by the more exacting requirements of pressure operation), than on comparable suction boilers.

Mr. Wingert also speculates on the "hazards and hardships placed on the operating crew as a result of pressure operation". Actually it has been found that many of the operators prefer pressure operation, basically because of improved stability of air flow control, and to date none of them have raised any fundamental objections to pressure operation.

Radioactive Cutting Tools for Rapid Tool-Life Testing

By M. E. MERCHANT,[1] HANS ERNST,[2] and E. J. KRABACHER,[3] CINCINNATI, OHIO

This paper presents an abbreviated method of measuring cutting-tool life using radioactive isotopes as tracers to measure the "instantaneous" rate of tool wear. This method consists of machining with a tool which has been rendered radioactive by neutron irradiation in a nuclear reactor, collecting the resulting chips, and measuring their radioactivity due to the particles abraded from the tool during a few seconds of cutting. The paper describes the preparation necessary, together with the equipment and techniques which were evolved in order to develop this test method. Data from tests conducted with various tool materials, work materials, rake angles, and cutting fluids are presented and analyzed. The results of special studies of the tool-wear mechanism also are presented.

Introduction

The vast economy of our country has been built on our ability to achieve great industrial productivity, the keystone of which may be found in our aptitude for mass production. This aptitude, in turn, is founded mainly on our ability and capacity for accurate high-production machining. The more economically such machining can be done, the higher will be our standard of living. Thus any means of lowering production machining costs is of direct concern to all. The most important mechanical factor influencing such costs, normally, is tool life (volume of metal removed per resharpening of the tool). Tool life enters machining costs in three ways—in down time for tool changes, in the labor and overhead involved in tool resharpening and repair, and in the replacement of worn-out tools. Thus any means of improving tool life results in a lower cost per piece and a higher production rate.

The search for increased tool life, through the development of improved cutting fluids, work materials, and machining conditions, requires controlled laboratory tool-life testing. The conventional laboratory method of tool-life testing usually consists of machining with a cutting tool, either until complete failure occurs or to a predetermined amount of flank wear, as measured with a microscope. In either case the test requires a great deal of time and material and involves considerable cost in order to obtain a relatively small amount of tool-life data. Attempts to accelerate these tests, as by employing abnormally high cutting speeds, usually yield results which do not agree with practice. In these tests the time required to obtain tool failure or the predetermined amount of flank wear determines the time required to run the test. It is quite apparent therefore that a method for evaluating tool life rapidly, using a smaller amount of material and at less cost, is quite desirable. It is also apparent that any such test method cannot be based on tool failure. It can, however, be based on the "rate" of tool wear if this can be determined by measurements made over a sufficiently short period of time.

The rate of wear of a cutting tool, when machining under normal conditions, is so low that to measure it by ordinary means, as by use of a microscope, requires observation over a long period of time, of the same order of magnitude as the life of the tool. This is illustrated in Fig. 1 where the observed wear on the flank of a cutting tool, as measured with a microscope, is plotted as a function of time. It can be seen that in order to establish with precision the slope of the resulting average straight line, and thus to determine the rate of wear, observations have to be made over a period of time nearly as long as that required to reach a flank wear of 0.030 in., which corresponds to tool failure. Thus only a small saving in the time required for tool-life testing can be obtained by measuring rate of wear if the ordinary means for making such measurements are used. Obviously, some extraordinarily sensitive means of measuring wear had to be found before a very rapid tool-life-testing method could be possible.

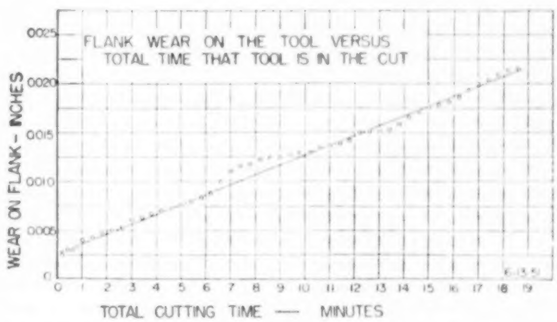

Fig. 1 Typical Relation Between Wear on Flank and Total Cutting Time

(SAE 8650 steel, sintered-carbide tool, 0 deg rake, 7 deg clearance, 591 fpm cutting speed, 0.0036 ipr feed.)

Following this line of thought, the idea was conceived of using a radioactive cutting tool to measure tool wear. It appeared that by this means it should be possible to determine the "instantaneous" rate of tool wear at any time throughout the life of the tool. This rate could be determined by measuring the amount of radioactivity of the collected particles worn from the tool during a given brief time of cutting, as by measuring the radioactivity of the mass of chips so produced. Since this amount of radioactivity would be directly proportional to the amount of metal worn from the radioactive tool during the given time, it should indicate directly the rate of wear of the cutting tool. Therefore, if the radioactivity of the collected wear products were measured for different cutting conditions, the relative rates of tool wear and thus the relative tool life for these different conditions would be made known. Thus this method appeared to have the required potentialities to supply the much

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needed rapid tool-life test. Therefore the method and its application were explored by the authors.

The Mechanics of Using Radioactive Tools

Theoretical Background. The "radioactive tool-wear method" selected for exploration was conceived to consist, in essence, of machining with a cutting tool which had been rendered radioactive by neutron irradiation in a nuclear reactor and measuring the radioactivity of the collected particles worn from the tool during a few seconds of cutting. If it could be assumed that most of these wear products would adhere to the chips, then this presumably could be done most conveniently by measuring the resulting radioactivity of the chips. The well-known linear relationship obtained with conventional tool-life testing methods, when plotting flank wear on the tool as a function of cutting time, Fig. 1, gave reasonable promise that a measure of the gross tool wear (total wear products abraded from the tool) would also result in a smooth curve when plotted as a function of cutting time. Whether or not the resulting curve would be linear was of no serious consequence, so long as there were no abrupt changes in slope, since a measure of the radioactivity of the tool-wear products, obtained from a cut of short duration (several seconds), would be a measure of the instantaneous rate of tool wear and thus a measure of the slope of the curve at that instant. From this it may be seen that if alternate cuts were made, for example, with two cutting fluids, the relative rate of tool wear with the two cutting fluids could be measured and thus the relative tool life predicted. Any effect of a gradual change in slope of the general curve could be eliminated by averaging the results of a few alternate runs. The fact that the actual curve later was found to be essentially linear was a further advantage, though not a necessity for success of the method.

To investigate the theoretical possibilities of the method, preliminary studies had to be made to determine whether irradiation of conventional tool materials would yield radioisotopes suitable for tool-wear studies. This had to take into account the half life of the radioisotope, the type of radiation, and the radiation energies in "Mev" (million electron volts.) Table 1 shows three of the tool materials selected for use in the investigation, together with their nominal chemical composition and the resulting effective tracer radioisotopes due to neutron irradiation in a nuclear reactor. Those radioisotopes which have sufficient half life and energy to render the tool useful for a considerable period of time are termed effective radioisotopes. Other radioisotopes are present after irradiation; however, they are either of such short half life that by the time the tips are received and ready for use the radioisotopes have decayed to the extent that their radioactivity may be neglected for purposes of this investigation (for example, W 187), or their activity is so slight as to be of negligible value in the test (for example, Fe 59 in high-speed steel). The half life of a radioisotope is the time required for the radioactivity of a given amount of the element to decay to half its initial value. Table 2 lists the half-life values for the effective radioisotopes.

The characteristics of the radiation emitted by these radioisotopes are also important to the success of the test method and are given in Table 2. Here it may be seen that both beta (β) and gamma (γ) radiation are present. Beta radiation consists of high-energy electrons. These beta particles possess slight penetrating power. Gamma radiation consists of high-energy, short-wavelength x rays. These rays have high penetrating power; they can penetrate several inches of lead. The fact that these two types of radiation are present is fortunate in that it allows either beta or gamma radiation to be measured in studying the tool-wear products. This factor affords the opportunity of making a variety of basic studies which would not be possible if only one of the types of radiation was present. It also may be seen from Table 2 that the energies of all beta radiations of the various radioisotopes are very nearly equal, as are the energies of the gamma radiations (with the exception of the chromium 51 isotope found only in the high-speed steel). This is a fortunate coincidence in that it offers a fairly homogeneous composite radiation with which to work. Thus the difference in decay rate

TABLE 1 COMPOSITION AND RESULTING RADIOISOTOPES FOR TOOL-TIP MATERIALS

TOOL MATERIAL	NOMINAL CHEMICAL COMPOSITION	RESULTING EFFECTIVE TRACER RADIOISOTOPES
Sintered Carbide (78)	WC 76%, Co 8%, TaC 4%, TiC 12% *	W 185, Ta 182, Co 60
Sintered Carbide (78B)	WC 82%, Co 10%, TiC 8% *	W 185, Co 60
H.S.S. (18-4-2)	C 0.8%, Mn 0.30%, Si 0.25% Cr 4%, V 2%, W 18%, Mo 0.75% Co 9%, Fe 64.9%	Cr 51, W 185, Co 60

* From "Properties of Carboloy Cemented Carbides," Carboloy Company, 1949.

TABLE 2 CHARACTERISTICS OF EFFECTIVE TRACER RADIOISOTOPES RESULTING FROM IRRADIATION OF VARIOUS TOOL MATERIALS

RADIOISOTOPE	HALF LIFE	PRINCIPAL RADIATION (Type & Energy)	Estimated Specific Activity of Tool Tip Material for 60 days irradiation (millicuries/gram)		
			SINTERED CARBIDE (78)	SINTERED CARBIDE (78B)	HIGH SPEED STEEL (18-4-2)
Co 60	5.2 Yrs.	B^- .31 Mev γ 1.33, 1.17 Mev	5.2	6.5	5.8
W 185	73 Days	B^- .43 Mev no γ	8.8	9.5	2.2
Ta 182	117 Days	B^- .53 Mev γ 1.2, 1.1 Mev	10.0	---	---
Cr 51	26 Days	no B (K capture) γ .32, .26 Mev	----	---	2.6
Total Estimated Specific Activity of tip material			24.0 mc/gm	16.0 mc/gm	10.6 mc/gm

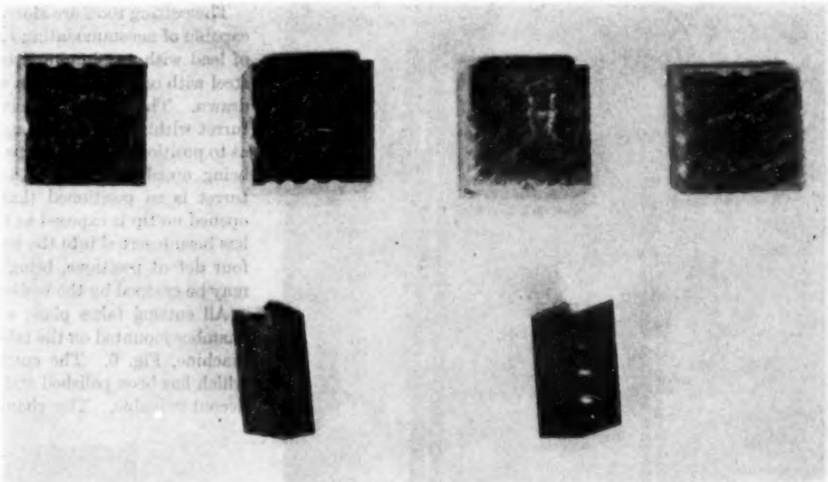

FIG. 2 TOOL BITS AFTER PREPARATION FOR IRRADIATION

(half life) of the various radioisotopes will have little effect on the characteristics of the over-all radiation.

Having established that the tool materials, upon irradiation, would furnish suitable radioisotopes for use in these tool-wear studies, the next step was to determine the required specific activity of these materials. The calculation of the required specific activity of the tool material was based on the premise that 50 disintegrations per sec could be counted without difficulty. Assuming, as average, an 80-min total tool life (cutting time) per cutting edge, and 6 sec cutting time per test cut (minimum), it was calculated that there would be approximately 0.3 microgram of tool-wear products per test cut. To provide the necessary 50 disintegrations per sec from this quantity of wear products requires a specific activity of 5 millicuries per gram for the tool tips. (1 curie = 3.7×10^{10} disintegrations per sec). From Table 2 it may be seen that the estimated specific activity of the three tool materials, after 60 days' irradiation, therefore, should be more than adequate for purposes of these wear studies. The calculations of the specific activity of the radioisotopes in Table 2 are based on the time of irradiation, the number of atoms per unit volume in the material being irradiated, usual average thermal neutron flux density of the Oak Ridge nuclear reactor, (5×10^{11} neutrons/cm^2/sec), the nuclear absorption cross section of the irradiated element, and the half life of the resulting radioisotope. The service irradiation of the cutting tools was carried out by the Oak Ridge National Laboratory on authorization from the Isotopes Division, U. S. Atomic Energy Commission.

The next field to be explored was that of the health physics of the operation. This subject is of utmost importance when doing any type of work with radioisotopes. Nuclear radiation must be understood and respected. It need not, however, be feared if proper precautions are taken. Every person is subject to small amounts of radiation at all times. Cosmic rays are constantly striking the earth everywhere. The air contains small amounts of radioactive carbon (C 14) to which we are subjected at all times. Naturally occurring radioisotopes present in the earth's crust give off radiations. These small amounts of radiation have no harmful effect on the human body. Similarly, small amounts of radiation received when working with radioisotopes in the laboratory have no harmful effects. There is, however, a definite limit to the amount of radiation to which one may be subjected without the possibility of eventually sustaining harmful effects. The National Committee on Radiation Protection, sponsored by the National Bureau of Standards, has established this tolerance limit (with a reasonably large factor of safety) as three tenths of one roentgen (0.3r) per week. One roentgen (r) is that quantity of x or gamma radiation which will produce one electrostatic unit of ions in a cubic centimeter of air under standard temperature and pressure. This amounts to a delivered energy of 83 ergs per gram of air.

When working with radioactive materials with a level of activity of the order of 10 to 20 millicuries, as with the tool tips used for this investigation, some protection is necessary and some precautions must be taken. Adequate protection from beta radiation may be obtained by use of light shielding. A sheet of transparent plastic $^1/_8$ in. thick will stop all beta particles of the energy of those given off by the tool materials, as listed in Table 2. Protection from gamma radiation is more difficult owing to the penetrating properties of the gamma rays. However, in addition to the protection afforded by direct shielding, very good protection is provided by controlling the distance between the worker and the radioactive material and by limiting the time of exposure.

Based on these health physics considerations, the test equipment and handling instruments for the radioactive tool-wear studies were so designed as to give more than adequate protection from beta and gamma radiation. This was accomplished by providing sufficient isolation of the tool tip throughout the entire test procedure, both by the use of shielding and by the use of remote handling equipment, as decribed later in the paper. As a result, the average dosage regularly received during the use of the radioactive tools in the investigation has been far below the tolerance level. The maximum amount of radiation received by a worker in any single week has been 0.011 roentgen. This is $^1/_{30}$ of the stated permissible tolerance. The average weekly dosage received by the worker receiving the greatest exposure has been approximately $^1/_{60}$ of the permissible tolerance. Thus it can be seen that the use of radioactive tools need involve no special hazards or harmful effects, any more than the use of electricity, high-pressure fluids, or harmful chemicals need do so when properly handled.

Experimental Equipment and Methods. The equipment and techniques required for the investigation of the radioactive tool-wear method were developed in accordance with the theoretical principles presented in the previous section. The cutting tools used for the tests are small tool bits approximately $^5/_{16}$ in. square $\times$ $^3/_{32}$ in. thick and weighing approximately 2 grams each. These are finish-ground to shape, and marked for identification before

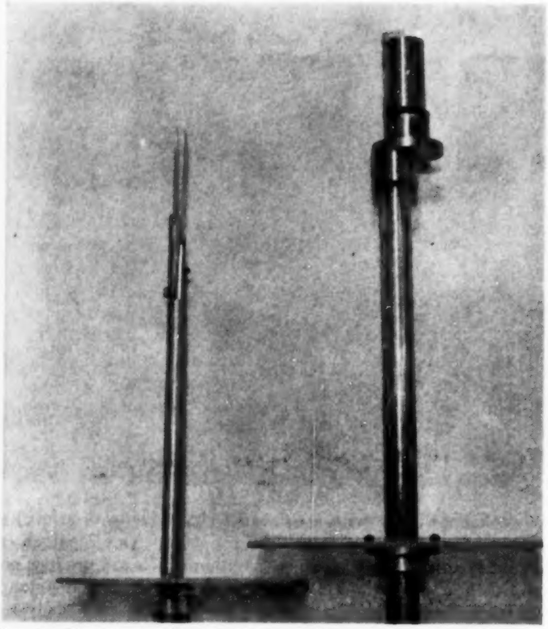

Fig. 3 Extension Tweezers and Toolholder for Handling Radioactive Tool Bits

they are irradiated. Fig. 2 shows several tool bits which have been prepared for irradiation. They are ground to a rhombohedral shape so that four cutting edges are available on each tip. The letter markings on the tips, Fig. 2, identify them as being of Stellite, high-speed steel, and Tantung. The small notches ground on the clearance edges make possible the identification of individual cutting edges, at a safe distance, after the tips have been irradiated.

Early studies, made with some of the first tips which were irradiated, indicated that considerable time could be saved and preliminary test runs eliminated if an initial 0.003 to 0.005-in. wear land was ground on the cutting edges before the tool tips were irradiated. This allows almost immediate access to the linear region of gross tool wear described later in this paper. After all four cutting edges have become dulled the tool tip is returned to Oak Ridge National Laboratory for disposal. It has been found to be more practical to dispose of the tips rather than to regrind them for further use because of their relatively high activity and the problem of handling resulting radioactive dust.

The fact that each individual tool-life determination requires only from 6 to 30 sec cutting time allows several hundred tests to be made with each tool tip. This fact more than offsets the cost of having a new tip irradiated.

All handling of the tips is done remotely owing to their radioactivity. This is accomplished by use of a pair of tweezers and a toolholder with long extension handles as shown in Fig. 3. The toolholder is so designed that it may be used to remove the tool tip from the storage chamber, and to hold it securely in position during the cutting operation. The toolholder with a tool tip in position is shown in Fig. 4. The toolholder consists primarily of the holder body and the clamping member. The holder body furnishes the seat upon which the tool tip is positioned. This seat is ground to give the desired tool-rake angle. The clamping member is spring-loaded to hold the tool tip in position during handling. Additional positive clamping is supplied during the cutting operation.

The cutting tools are stored in a lead and steel chamber, Fig. 5, capable of accommodating four tool tips. The outer chamber is of lead with an aluminum lining, while the inner chamber is of steel with one port through which the cutting tools may be withdrawn. The tools themselves rest on individual shelves of a turret within the inner chamber. The turret may be rotated so as to position any of the tips for withdrawal without the operator being openly exposed to any direct radiation. Normally the turret is so positioned that when the outer chamber door is opened no tip is exposed to the open port. After the toolholder has been inserted into the port the turret is rotated to one of the four detent positions, bringing a tool tip into position where it may be grasped by the toolholder and withdrawn.

All cutting takes place within a completely enclosed cutting chamber mounted on the table of a knee-and-column-type milling machine, Fig. 6. The cutting chamber is constructed of steel which has been polished and chromium-plated to render it easily decontaminable. The chamber walls are of sufficient mass to

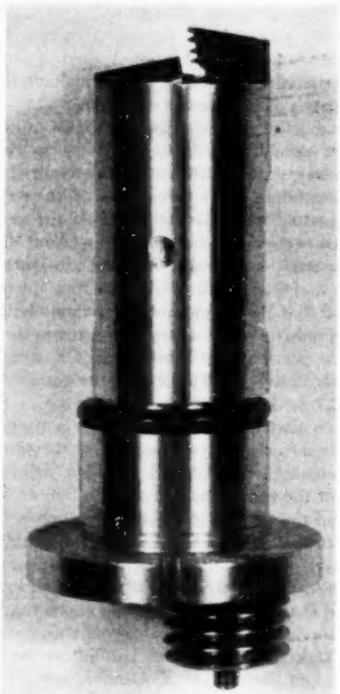

Fig. 4 Toolholder With Tool Bit in Position

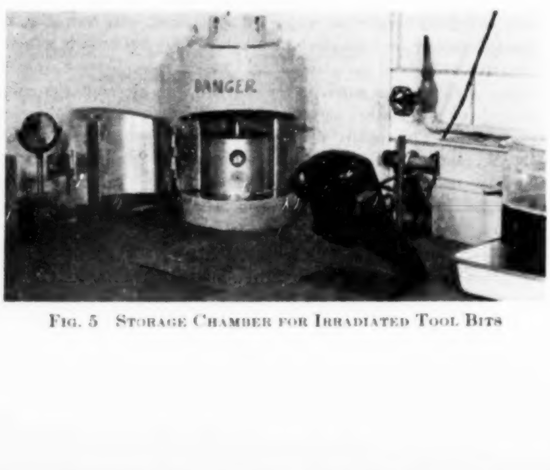

Fig. 5 Storage Chamber for Irradiated Tool Bits

protect the operator completely from any harmful amounts of radiation. The chamber contains its own cutting-fluid circulating system which supplies cutting fluid to the tool from two nozzles, one on either side of the cutting tool. The cutting fluid is returned, by means of small slots, to the reservoir which is immediately beneath the cutting chamber.

A tubular workpiece is used. It is held in a draw collet adapted to the spindle of the milling machine, and enters the cutting chamber from the rear, through a seal, Fig. 7. The cutting tests

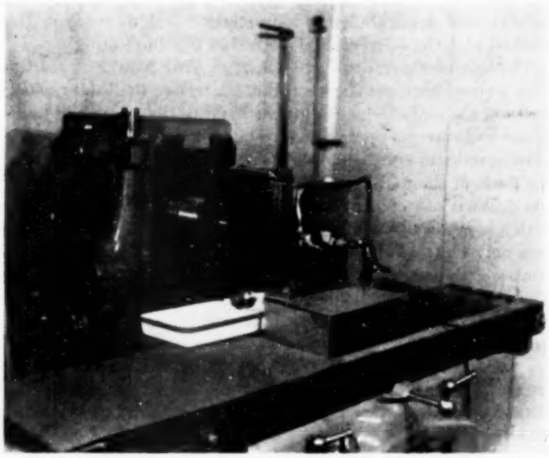

Fig. 6 General View of Cutting Chamber and Setup

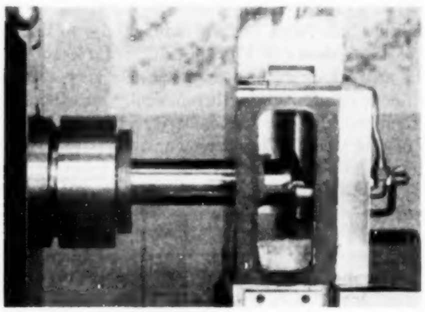

Fig. 7 Interior of Cutting Chamber

are made by inserting the toolholder with the cutting tool into the cutting chamber, clamping it securely, and cutting on the end of the tubular workpiece. A wide range of feeds and speeds are available to give desired cutting conditions. After the cut has been made the toolholder and tool are withdrawn from the chamber and stored in a special storage block for convenient shielding, while the resulting chips which have been collected in a stainless-steel wire-mesh basket are removed through the cutting-chamber door. The chips are then cleaned, dried, and weighed preparatory to counting. A right-angled microscope setup, provided with a micrometer eyepiece, is used for inspecting the cutting edge of the tool and for measuring flank wear when desired. The microscope is mounted in a steel block to provide proper shielding from radiation. This setup is shown in Fig. 8.

The laboratory, in general, is constructed and equipped in accordance with suggested practice for radioisotope work. The flooring is of heavily waxed plastic tile. The sink is of stainless steel. All bench tops and the interior of the hood are painted

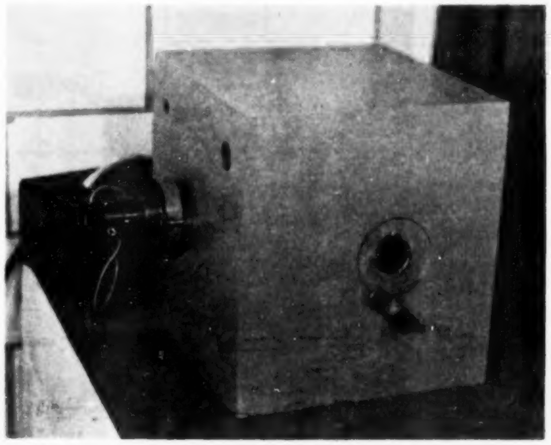

Fig. 8 Microscope Setup for Flank-Wear Measurements on Radioactive Tools

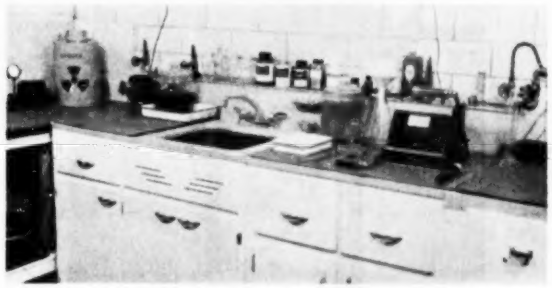

Fig. 9 View of Laboratory Work Area

with a strippable plastic coating. In addition, the bench tops are covered with plate glass and absorbent paper. The hood has a forced draft in excess of 100 fpm with the front full open. All discharge air from the hood is filtered. The hood is used for all wet work and any work where there is apt to be contaminated dust or small particles. All work in the hood is handled in stainless-steel trays designed to catch possible spills. Radiation monitoring equipment is also available in the laboratory for personnel protection and for the monitoring of sources of possible contamination. A general view of part of the laboratory work area is shown in Fig. 9. Stringent rules are enforced in the laboratory at all times for personnel safety. All persons working in the laboratory must wear protective clothing, film badges, and dosimeters. No smoking or eating is permitted. Rubber gloves are a must when handling anything suspected of being contaminated. Personnel cleanup and monitoring are required before leaving the "hot lab."

Measurement of the activity of the chips for determining the instantaneous rate of tool wear is carried out in a part of the laboratory away from the hot lab, where the background readings are normal. Fig. 10 shows a general view of the automatic scaler, the counting chamber, which is a lead chamber, aluminum lined, used for end-window beta counting, and a heavy walled gamma tube. Both beta and gamma-counting techniques have been investigated as a means for measuring the activity of the wear products present on the chips. Gamma counting was found to be quite satisfactory for the wear-rate determinations. The beta-counting technique has been used for special studies of individual chips.

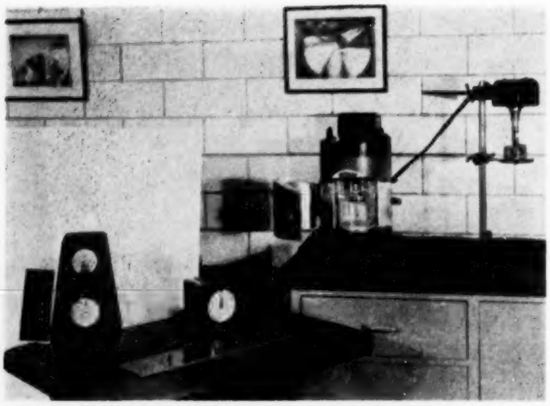

Fig. 10 General View of Scaler and Counting Setup

Fig. 11 View of Gamma-Tube Setup for Counting Chips

For wear-rate measurements the chips are placed in a Marinelli beaker for counting. This type of beaker contains a central cylinder into which the Geiger tube may be inserted for gamma counting without coming in contact with the chips. A close-up view of the setup is shown in Fig. 11. The automatic scaling circuit may be set to count for a predetermined number of counts before it shuts off. The number of predetermined counts chosen is determined by the desired counting accuracy. The normal time required for counting to a $\pm 2^1/_2$ per cent accuracy is from 2 to 4 min. The total counts are divided by the counting time to give counts per minute (gross). The background counts per minute (counts per minute recorded by the scaler due to stray radiation only, no source being present) are subtracted from this value to give counts per minute (net). This value, in turn, is divided by the weight of chips in grams to give the final value of counts per minute per gram of chips (cpm/gram) which is a direct measure of the rate of tool wear.

The experimental test methods and equipment used to date in this investigation, and described herein, are by no means the only way of applying radioactive cutting tools to the study of tool wear. There is no reason, for example, why this method would not be just as applicable to a lathe-turning operation if proper shielding and other necessary precautions were employed.

Tool-Life Testing

Validity of the Method. The method of tool-life testing presented in this paper, in order to be of value to the practice of metal cutting, must be more than a rapid method of studying tool wear. It must be applicable to the study of relative tool life when evaluating various cutting fluids, work materials, tool materials, and other varying cutting conditions. Data obtained by this method also must be at least as reliable and reproducible as those obtained by accepted methods. The data presented in the following sections of this paper are given to establish the validity and applicability of this method and to compare this method with the conventional method of tool-life testing.

The results of a series of cuts taken on AISI 8650 steel, starting with a sharp tool, are shown in Fig. 12. Here the total radioactivity of the chips (which is proportional to the amount of radioactive tool material present on the chips), as accumulated throughout the series of cuts, is plotted against tool wear as measured on the flank of the tool with a microscope in the conventional way. (As pointed out earlier, the activity of the chips was measured with a heavy-wall Geiger tube and thus represents gamma-radiation only.) It can be seen that the measured radioactivity correlates with the wear measurements made with a microscope. The scattering of the experimental points, evident in this case, is due not nearly as much to variations in the radioactivity measurements as to the "cascading" of the gross wear on the flank (as measured with the microscope). This can be seen by reference to Fig. 1, which is a plot of the actual flank-wear data obtained during this same series of cuts. The cascading effect evident there is well known and is one of the important factors which often prevent obtaining reliable measurements of the rate of tool wear from short-time tests, covering only a small portion of the wear curve, when using conventional methods.

On the other hand, a plot of the radioactivity of the chips as a function of time, shown in Fig. 13, exhibits (after initial "break-in" of the sharp tool during which wear is very rapid), an essentially linear relationship with practically no cascading. It is evident therefore that the radioactivity of the chips is a valid measure of tool wear. Thus an accurate measure of the rate of tool wear can be obtained by averaging the values for the activ-

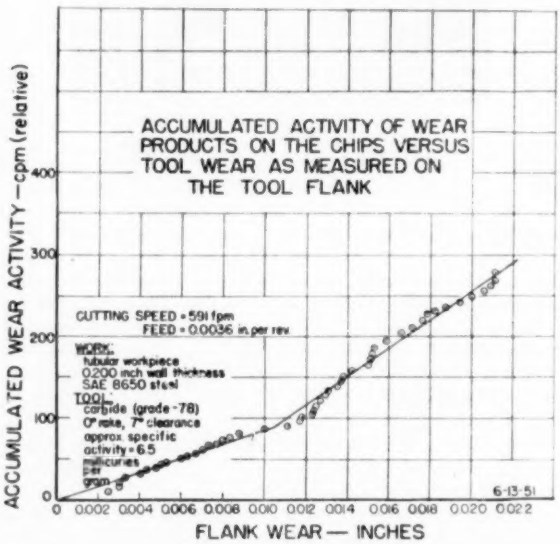

Fig. 12 Relation Between Accumulated Radioactivity on Chips and Flank Wear

ity of the chips (cpm/gram of chips) obtained from only a few runs of a few seconds' duration each.

The foregoing findings raise a question as to the relative amount of the total wear products that remains with the chips as compared to the amount that accumulates in the cutting fluid. If a large and variable portion of the wear products goes into the cutting fluid rather than remains with the chips, measurement of the radioactivity of the chips alone would not be sufficient. To answer this question, a study was made of the type and amount of wear products present in the cutting fluid. The cutting-fluid

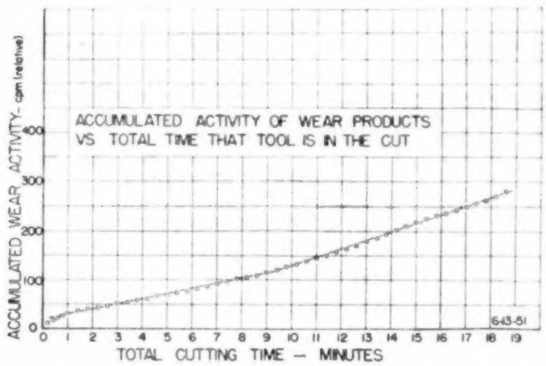

FIG. 13 RELATION BETWEEN ACCUMULATED RADIOACTIVITY ON CHIPS AND TOTAL CUTTING TIME; 591 FPM

samples were counted in the same manner as were the chip samples (with the heavy-walled gamma tube). A 150-cc sample of cutting fluid was taken as being representative of the total 1000-cc quantity of fluid in the circulating system.

The cutting fluid was allowed to circulate for some time before a sample was taken, in order that it be as representative as possible. However, it was found that the general wear products in the cutting fluid were so fine that they were not even removed by filtering the cutting fluid through ordinary filter paper. This indicates that the wear products in the cutting fluid are either very fine particles in suspension or are actually in solution. Occasional larger particles which slough off the tool once in a while and remain on the bottom of the cutting-fluid reservoir were not considered to be a true wear product and therefore were neglected in these considerations. Fig. 14 is a plot of the accumulated activity of the wear products collected in the 150-cc sample of cutting fluid as a function of the total time that the tool was in the cut, for this same series of tests. It can be seen that after a short break-in period the relationship is fairly linear.

From a comparison of Figs. 13 and 14, the percentage of the total wear products which adhere to the chips may be calculated. For a given total cutting time, say, 15 min, the total accumulation of wear products in the 150-cc sample of cutting fluid (in terms of its observed radioactivity) is found, from Fig. 14, to be 295 cpm. The total amount of wear products in the 1000-cc volume of cutting fluid circulated would therefore be $295 \times 1000/150 = 1965$ cpm. The total amount of wear products accumulated by the chips may be obtained from Fig. 13 by multiplying the accumulated wear activity given there by 100, since the actual counts were taken on the full 100 grams of chips removed in each cut and then reduced to a 1-gram basis for plotting.

Thus from Fig. 13 the total accumulated wear activity on the chips for 15-min total cutting time is found to be 22,000 cpm. This value must in turn be doubled since only the wear activity of the chips for alternate cuts was measured and plotted. The total accumulated wear activity that adhered to the chips for the 15-min total cutting time would, therefore, be 44,000 cpm. From the foregoing data, the percentage of the total tool-wear products that adhere to the chips may be calculated as being $44{,}000/(44{,}000 + 1965) = 95.8$ per cent. Since this method of determining the percentage of the total wear products which adhere to the chips admittedly involves several approximations which make this calculated figure lower than the actual value,

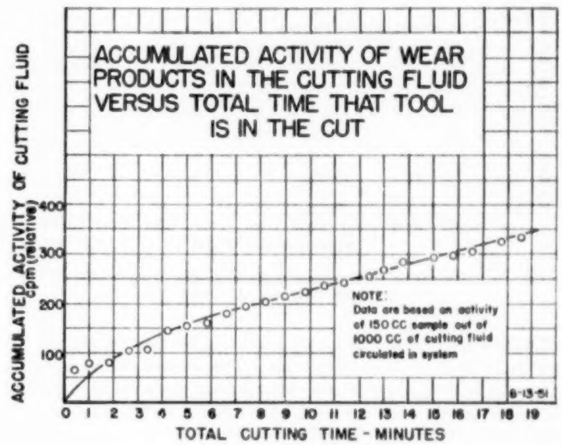

FIG. 14 RELATION BETWEEN ACCUMULATED RADIOACTIVITY IN CUTTING FLUID AND TOTAL CUTTING TIME; 591 FPM

it is quite evident from the example given that well over 90 per cent of the wear products remain with the chips. This being the case, it is valid to assume that the rate of tool wear is established when the activity of the chips has been measured.

Figs. 15 and 16 show the results of another series of runs made under conditions identical with those in Figs. 1, 12, and 13, except for the cutting speed. Comparison of Figs. 12 and 15 and of Figs. 13 and 16 shows the similarity of the results obtained at the two cutting speeds. Comparison of Figs. 13 and 16 shows, however, that for a given time of cutting, the accumulated wear activity is considerably less at 347 fpm than at 591 fpm, as would be expected from the usual relationship between cutting speed and tool life. That these data do conform strictly to that usual relationship is borne out in Fig. 17 which shows a plot of tool wear, as measured by the radioactivity of the chips, as a function of cutting speed. Here average data are plotted for cutting speeds of 347, 457, and 591 fpm. It may be seen that the tool-wear values, which are a direct measure of tool life, follow a linear relationship on the log-log plot exactly similar to the well-established linear log-log relationship between tool life and cutting speed as obtained from conventional tool-life-testing methods. Thus it appears that the same empirical relationship which has been well established for conventional tool-life-testing methods also exists for this new rapid method of tool-life testing.

Typical Test Results. Although the radioactive tool-wear method has been in use only a relatively short time for obtaining data for application to production-machining operations, numerous practical tool-life studies already have been made with it. The results of a few such studies, typical of some of the types of problems to which the method is applicable, are presented here to illustrate its use.

The data presented in Table 3 are results obtained from an investigation of the relative tool life obtainable with two cutting fluids. This information was needed to determine the relative merits of the two formulations for use in practice. Three series

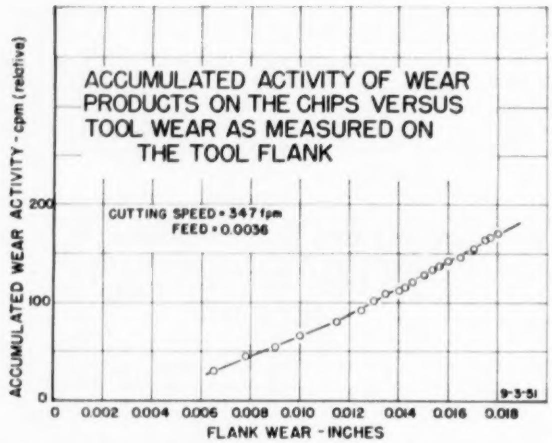

FIG. 15 RELATION BETWEEN ACCUMULATED RADIOACTIVITY ON CHIPS AND FLANK WEAR

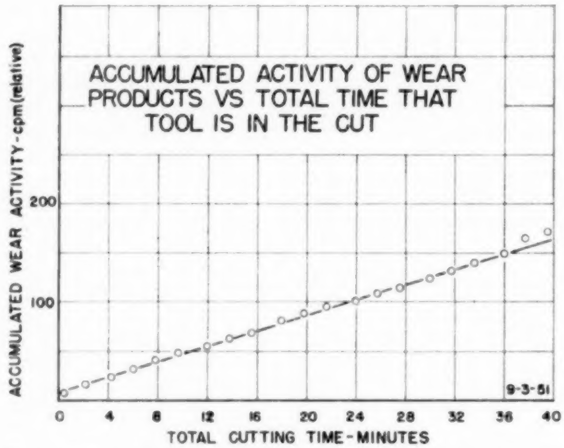

FIG. 16 RELATION BETWEEN ACCUMULATED RADIOACTIVITY ON CHIPS AND TOTAL CUTTING TIME; 347 FPM

of tests were made, each series consisting of three individual 12-sec runs with each cutting fluid. The cutting fluids were alternated after each set of three runs in order to minimize any effect of variation in work material or condition of the cutting tool. The radioactivity of the chips was measured by the afore-mentioned method to determine the cpm/gram of chips, which is a measure of the rate of tool wear. This in turn is inversely related to expected tool life.

The results obtained from each individual test are shown in Table 3. The average value presented is the average of the nine individual test runs. From these data it may be seen that the average amount of wear products present per gram of chips is approximately 11 per cent less for the tests made with cutting fluid B than for those with A. This would indicate that for these test conditions cutting fluid B may be expected to give approximately 11 per cent greater tool life than cutting fluid A.

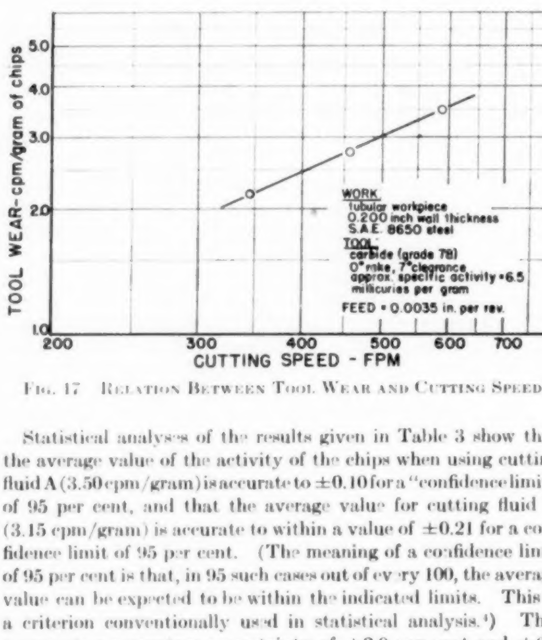

FIG. 17 RELATION BETWEEN TOOL WEAR AND CUTTING SPEED

TABLE 3 TYPICAL RELATIVE TOOL-LIFE EVALUATION—TWO CUTTING FLUIDS

Workpiece: S. A. E. 8650 steel (tubular)
Width of cut: 0.2000 in.
Feed per revolution: 0.0036
Cutting Speed: 591 fpm
Tool: Sintered Carbide (78) 0° rake 7° relief

	CUTTING FLUID	
	"A"	"B"
Counts per minute net above background, per gram of chips, due to tool wear products	3.65	3.71
	3.54	3.12
	3.49	2.80
	3.40	3.26
	3.50	2.88
	3.79	2.94
	3.34	3.50
	3.55	2.88
	3.23	3.23
Average	3.50 ± 0.10	3.15 ± 0.21
Uncertainty	± 2.9%	± 6.7%
Relative Tool Life	100%	111%

Statistical analyses of the results given in Table 3 show that the average value of the activity of the chips when using cutting fluid A (3.50 cpm/gram) is accurate to ±0.10 for a "confidence limit" of 95 per cent, and that the average value for cutting fluid B (3.15 cpm/gram) is accurate to within a value of ±0.21 for a confidence limit of 95 per cent. (The meaning of a confidence limit of 95 per cent is that, in 95 such cases out of every 100, the average value can be expected to be within the indicated limits. This is a criterion conventionally used in statistical analysis.[4]) This represents a percentage uncertainty of ±2.9 per cent and ±6.7 per cent, respectively, as seen from Table 3. From this it may be seen that the difference in tool life of 11 per cent between the two cutting fluids is quite significant. Thus, from the foregoing example, the radioactive tool-wear method seems to be well suited to the measurement of small significant differences in tool life obtainable with two different cutting fluids.

Table 4 presents the results of an investigation conducted to determine the relative tool life obtained when machining three approximately equivalent AISI steels, these three steels being AISI 3145, AISI 8650, and AISI 80B-45. This information was needed for use in determining relative cutting speeds for these three steels for production-machining operations. Each of the

[4] "ASTM Manual on Quality Control of Materials," by ASTM Committee E-11, American Society for Testing Materials, January, 1951.

three steels was given the same standard heat-treatment, duplicating the condition in which it would be machined in practice.

For this investigation a grade 78 B sintered-carbide tool was used with a +10-deg rake and 7-deg relief. The cutting fluid was a standard chemical-emulsion type. Each test cut was of 0.5 min duration. The three work materials were used in rotation, changing after each cut, in order to minimize any variation in the condition of the cutting tool. The results obtained from each individual test are shown in Table 4. The average value presented is the average of the eight individual test runs. These values are converted to relative tool-life values, taking the tool life obtainable with the AISI 3145 as 100 per cent.

The results of this investigation indicate that there is no significant difference in tool life for the three steels since the plus or minus uncertainty for a 95 per cent confidence limit is sufficient in each case to cause overlapping of the average values. It should be pointed out that while no significant difference in tool

TABLE 4 TYPICAL RELATIVE TOOL-LIFE EVALUATION—THREE WORK MATERIALS

Workpiece: A.I.S.I. 3145, A.I.S.I. 8650 and A.I.S.I. 80B-45 steel (tubular)
Width of cut: 0.200 in. Cutting Fluid: Water Soluble (1:20)
Feed per revolution: 0.0035 Tool: Sintered Carbide (78B)
Cutting speed: 145 fpm + 10° rake, 7° relief

MATERIAL	A.I.S.I. 3145	A.I.S.I. 8650	A.I.S.I. 80B-45
Counts per minute net above background, per gram of chips, due to tool wear products.	16.7	17.6	20.8
	17.3	17.9	20.8
	15.5	18.7	16.8
	15.9	16.9	17.2
	15.0	19.3	18.5
	16.0	18.8	18.4
	18.1	13.4	14.9
	18.1	17.6	20.8
Average	16.58 ± .91	17.53 ± 1.5	18.53 ± 1.8
Uncertainty	± 5%	± 8%	± 10%
Relative tool life	100%	94%	89%

life was found in this case, the test method, nevertheless, is well suited to the study of relative tool life obtainable for various work materials. Had the difference in tool life obtainable (machinability) been greater than 10 per cent for any of the three materials the resulting difference would have been significant. However, since these three particular steels, when given similar heat-treatments, are practically equivalents, the finding that they all gave essentially similar tool life was in line with expectations.

An investigation was conducted to determine the relative tool life obtainable with two different cutting fluids when machining aluminum with a high-speed-steel tool. This information was needed to determine whether the two equivalent formulations, for use on a production-machining operation on aluminum, might be expected to give any different performance. The cutting time for each test was 6 sec. The data obtained and the test conditions are given in Table 5.

For this series of tests the cutting fluids were alternated after each test rather than after every three runs as was the case for the data in Table 3. This procedure has now become the practice when evaluating the relative tool life obtainable with two cutting fluids, in order to eliminate as fully as possible the effect of local variations in the work material and gradual changes in the cutting tools. The general practice, at present, is to run tests with each

TABLE 5 TYPICAL RELATIVE TOOL-LIFE EVALUATION—TWO CUTTING FLUIDS

Workpiece: Aluminum (tubular)
Width of cut: 0.200 in. Tool: H.S.S. (18-4-2)
Feed per revolution: 0.0036 + 10° rake
Cutting Speed: 755 rpm 7° relief

	CUTTING FLUID "C"	CUTTING FLUID "D"	Ratio of C/D
Counts per minute net above background, per gram of chips, due to tool wear products.	2.51	2.96	85
	1.72	1.95	.88
	1.60	1.49	1.07
	1.36	1.55	88
	1.84	1.89	.97
	1.65	1.60	1.03
	1.31	1.26	1.04
Average			96 ± .08
Uncertainty			± 8.3%
Relative Tool Life	104%	100%	

cutting fluid individually, making alternate tests between it and a "standard" cutting fluid. The relative tool life obtained with this particular cutting fluid as compared to the standard may in this way be compared with that of other cutting fluids at any time in the future.

Another difference is found in the interpretation of the test data, as may be seen from Table 5. In this case the individual ratios of the wear values for cutting fluid C to those for cutting fluid D are averaged as the basis for comparison. While this method has no effect on the final average relative value, it does give a truer picture as to the plus or minus statistical uncertainty. The data in Table 5 indicate the expected tool life with cutting fluid C to be 4 per cent greater than that with D. However, the statistical uncertainty is found to be ±8.3 per cent, therefore indicating that in the case of these two particular cutting fluids there should be no significant difference in tool life in practice.

The foregoing data demonstrate that this method is as well adapted to the use of an aluminum workpiece and a high-speed-steel tool of +10-deg rake as it is to the use of a steel workpiece using a carbide tool of 0-deg rake, when making relative tool-life evaluations for various cutting fluids.

These samples of typical data were presented, as previously mentioned, to establish the applicability of this method to tool-life evaluations under various cutting conditions, such as with different work materials, tool materials, and cutting fluids. Many additional data, not given here, have been obtained for numerous series of tests using this test method to provide tool-life data needed for the solution of various practical problems.

The reliability of the tool-life data obtained by this new method of tool-life testing is established in its kindred relationship to the conventional tool-life-testing method as pointed out in connection with Figs. 12, 15, and 17. The repeatability of this method, and its ability to show relatively small differences in tool life accurately are substantiated by calculating the data on the basis of a 95 per cent confidence limit. As may be seen from Tables 3, 4, and 5, the normal plus or minus uncertainty to be expected lies in a range between 5 and 10 per cent. If this amount of uncertainty seems a little high, let us look at tool-life data obtained under carefully controlled conditions by means of the conventional tool-life-testing method. For this method the tool life was determined by the time required to obtain 0.030-in.

TABLE 6 STATISTICAL UNCERTAINTY FOR TYPICAL CONVENTIONAL TOOL-LIFE TESTS

Tool: H.S.S. 2.5 - 9.5 - 5 - 5 - 5 15 - 1/32
Depth of Cut: 0.080 in.
Feed per revolution: 0.011 in.

Cutting Speed fpm.	Flank Wear in.	No. of Test runs averaged	Average Tool Life (min.) for 95% confidence limit	Percent Uncertainty based on average for 95% confidence limit.
50	030	6	77 ± 25.6	± 33.3
50	030	9	48.78 ± 15.7	± 32.2
75	.030	6	39.42 ± 23.4	± 59.0
75	.030	8	31.25 ± 9.1	± 29.0
75	.030	7	28.7 ± 10.2	± 35.5
120	.030	10	5.1 ± 1.6	± 31.2
			Average Uncertainty ± 36.7%	

flank wear on the cutting tool—a common method of tool-life testing.

Table 6 shows these data. Here the tool life in minutes was averaged for a number of test runs, as indicated, and the per cent uncertainty figured on the basis of a 95 per cent confidence limit, as was done for the radioactive tool-wear method. It can be seen from Table 6 that the average plus or minus uncertainty is 36.7 per cent for these six series of tests. This is approximately 3 to 5 times the amount of uncertainty obtained with the radioactive tool-wear method. Yet in making the conventional tool-life tests, every precaution was taken to eliminate possible causes of variation.

A comparison may be made from Table 6 as to the relative cutting time required to conduct a similar series of tool-life tests by the conventional method, as in Table 6, and by the radioactive tool-wear method. As an example we may use the data from that series of tests from Table 6 for which the cutting speed was 75 fpm and the average tool life was 31.25 min. For the 8 runs made in this case the total cutting time was 250 min. The cutting time required to make 8 tool-life determinations by the radioactive tool-wear method is only $^1/_{25}$ that for the conventional tool-life-testing method, while the expected plus or minus uncertainty from the average for a 95 per cent confidence limit would be much less than the resulting 29 per cent for the series of tests of Table 6.

In addition to the total cutting time involved in making the tests with the radioactive tool-wear method, there would of course be time involved in handling and in the radioactivity measurements. However, this is well offset by the time involved in tool grinding and setup with the conventional tool-life-testing method, so it is obvious that the radioactive tool method is a tremendous timesaver.

Special Studies

The application of radioisotopes to the fundamental study of the tool-wear process should add considerably to our limited knowledge of the mechanics of this process. While it is true that general studies of tool wear have been made for a long time, nevertheless the fundamental happenings accompanying this wear process have until now remained beyond our reach—behind a closed door. Radioisotopes would appear to be the key needed to open this portal to a fundamental understanding of this process. While no extensive studies along this line have been made by the authors to date, vast possibilities are recognized.

A few preliminary findings can be mentioned at this time. Fig. 18 is an autoradiograph of an AISI 8650 steel chip turned with an irradiated high-speed-steel tool at 147 fpm. The feed per revolution was 0.0035 in., rake +10 deg, relief 7 deg. The time of exposure for this autoradiograph was 43 days; it was made from the side of the chip which had contacted the face of the cutting tool. This picture, which clearly delineates the chip, was produced entirely by the radiation coming from the tiny particles of radioactive tool material (wear products) adhering to the chip. It shows the quite uniform distribution of these wear products along the chip. The wear rate in this case was rather high. Other autoradiographs made from chips obtained under conditions where the rate of wear was less, though not as clear as this, show an even finer and more uniform distribution of wear products. The autoradiographs show that the tool wears by a process of continuous abrasion. Studies of this nature, when more fully explored, should yield much useful information.

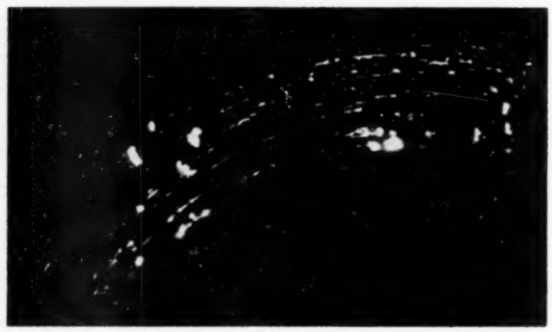

Fig. 18 Chip Autoradiograph

As pointed out earlier in this paper, it has been found, by use of radioisotopes, that very nearly all of the tool-wear products adhere to the chips. This fact, unknown until now, resulted from a very simple study using the radioactive tool material as a tracer. Studies along this line should contribute much toward a better basic understanding of the tool-wear process.

Another study, practically impossible to make by conventional methods, is conveniently made using the tool-wear products as a radioactive tracer. Here use is made of the beta radiations to study the relative quantities of tool material worn from the tool face and tool flank. By counting the beta radiation only, the amount of tool-wear products on each side of the chip may be counted separately. This is possible because beta particles will not penetrate the chip. Preliminary measurements of this type indicate that the face wear and flank wear are roughly equal.

The foregoing examples are only indications of the extreme usefulness of radioisotopes in wear studies of this type. Through their use the avenues have been opened to a clearer and more complete understanding of the mechanics of the process of tool wear.

Conclusions

While the radioactive tool-wear method of tool-life testing presented in this paper is still in an exploratory stage, leaving much room for improvement in techniques and thus in results, it has been found to function very successfully and to give results which are just as reliable and more reproducible than those obtained with conventional tool-life-testing methods. This is achieved at a vast saving in time and work material.

This method also opens new avenues for basic studies of the tool-wear process which heretofore have been inaccessible.

It is evident that the method is especially well-suited to the rapid determination of relative values of tool wear and tool life when evaluating the effect of a series of different cutting conditions, as, for instance, a series of different cutting fluids or different work materials. This method thus offers considerable promise as a means of greatly speeding up the collection of tool-life data in the laboratory, thereby allowing more complete and more rapid evaluation of advances in metal-cutting science and practice.

Acknowledgments

The authors wish to express their appreciation to Mr. James Howard, formerly of this company, for his help in the design of equipment, and to Mr. George Hain of the Research Department, for his aid in establishing test methods and in the building of special test equipment. Thanks are also due Mr. Karl Westerlund, formerly of this company, who redesigned and rebuilt the automatic scaling circuit, and Miss Sue Hannaford and Mrs. Margaret Huenefeld of the Research Department, for their help in the preparation of the illustrations and manuscript.

The Influence of Higher Rake Angles on Performance in Milling

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This paper presents the results of research on the influence of rake angle on metal-cutting performance when using high-speed-steel cutters. A range of rake angles and cutting speeds was selected and studies made on cutter life and surface finish as a function of these variables. The effect of these variables on such basic quantities as forces, shear angle, chip friction, work done in cutting, and the calculated temperature on the chip-tool interface also was investigated. In several cases, excellent performance was obtained using higher rake angles (of the order of 25 deg to 40 deg) at higher cutting speeds. This observed performance is analyzed in relation to the basic factors.

Introduction

RAKE angles on the majority of high-speed-steel cutters in use today vary between 5 and 15 deg. The need for standardization of cutters and minimum inventories has played a certain part in keeping this value fixed. However, too often the only justification for the use of such angles is "tradition." Whether or not such angles are the best to use for a particular set of machining conditions is seldom known; if these angles have worked well in the past, they usually are assumed to be satisfactory for the present and future.

Such an approach leaves many questions unanswered, however. Could better performance be realized from the same high-speed-steel cutter merely by changing the rake angles? Could an increase in production be obtained with, at the same time, better finish and less power consumption?

Considerable research has been carried out on the influence of rake angles on tool life for single-point tools, especially for turning operations. The effect of cutter angles on tool life and general performance also has been well explored for the case of milling cutters tipped with sintered carbide. However, very little information has been available, until recently, regarding the influence of rake angles on tool life for milling with high-speed-steel cutters.

Among the earliest references to the advantages of increasing rake angles on high-speed-steel milling cutters to values beyond those commonly used are papers by Sinawski[3] and Frazer.[4] Both of these give figures taken from numerous production jobs supporting benefits resulting from use of high-speed-steel milling cutters having rake angles higher than normal, when applied to certain operations. However, no investigation appears to have been made, in the past, of the actual effect on cutter life and general performance produced by varying the rake angles of given high-speed-steel milling cutters over a wide range, under constant and controlled laboratory conditions, to determine optimum values.

Because of the foregoing facts, an investigation was launched by the authors to explore these fields. A face-milling and a slab-milling operation were chosen for study, as it was felt that these two would be representative of the majority of milling operations. In conjunction with the face and slab-milling operations, to provide an analysis of the mechanics of cutting with higher rake angles, orthogonal cutting tests were performed with a tool dynamometer. A range of rake angles from 0 to 50 deg was chosen for exploration, in conjunction with four general levels of cutting speed: 70, 160, 300, and 500 fpm. SAE 1020 hot-rolled stock was selected because it would serve as a more common index of milling performance than some of the higher-alloy steels. The work material was cut into sections 2 in. × 8 in. × 18 in. and ground to a uniform thickness to facilitate clamping and, more important, to remove the scale often found to be harmful to high-rake milling operations. This same material was used for both the face-milling and the slab-milling operations, and tubes also were fabricated from these blocks for use in the orthogonal cutting tests. A chemical emulsion-type cutting fluid, mixed 25 to 1, was used throughout these tests.

In the face-milling and slab-milling operations, studies were made on cutter life and surface finish as a function of the two variables, speed and rake angle. In the tool-dynamometer tests, evaluations were made of the influence of rake angle on such basic quantities as forces, shear angle, chip friction, work done in cutting and tool-chip contact area. From these data, tool-chip interface-temperature calculations also were made according to techniques developed by Trigger and Chao.[5]

The basic rake angle used as the variable in both the milling tests and the tool-dynamometer tests was the "true rake." This is the angle between the tool face and a plane perpendicular to the direction of relative motion of tool and work as measured in a plane containing this direction and perpendicular to the surface generated by the main cutting edge of the tooth or cutting tool. In the case of the milling operations, this true rake is a function of the axial and radial rake of the cutter and its corner angle. The relationship, as arrived at by Kronenberg,[6] is as follows:

$$\tan t = \tan r \cos c + \tan a \sin c$$

where

t = true rake angle of milling-cutter tooth, deg
a = axial rake angle, deg
r = radial rake angle, deg
c = corner angle, deg

In the case of the face-milling cutter, the values of the axial rake and radial rake were made equal, and a corner angle of 45

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[2] Assistant Director of Research, The Cincinnati Milling Machine Company. Mem. ASME.

[3] "HSS, Plus High Angles, Mills Like Carbide," by W. F. Sinawski, *American Machinist*, vol. 94, April 3, 1950, pp. 122–123.

[4] "Raising the Limits of High Speed Steel Machining," by W. R. Frazer, *Tool Engineer*, vol. 26, March, 1951, pp. 33–37.

Contributed by the Production Engineering Division, Metal Cutting Data and Bibliography Research Committee, and Cutting Fluids Research Committee and presented at the Semi-Annual Meeting, Cincinnati, Ohio, June 15–19, 1952, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, August 14, 1952.

[5] "An Analytical Evaluation of Metal-Cutting Temperatures," by K. J. Trigger and B. T. Chao, Trans. ASME, vol. 73, 1951, pp. 57–68.

[6] "Cutting-Angle Relationships on Metal-Cutting Tools," by M. Kronenberg, *Mechanical Engineering*, vol. 65, 1943, pp. 901–904.

deg was used, so that the main cutting edge of the cutter tooth was always perpendicular to the direction of relative motion of tooth and work (i.e., 0 deg inclination of the cutting edge). In the case of slab milling, the corner angle is always 0 deg, so that the true rake is equal to the radial rake, as may be seen from the foregoing equation. Thus this was so in the case of the present slab-milling cutter also, but since it had a helix angle of 45 deg, the cutting edge was inclined at 45 deg to the direction of relative motion of tooth and work (i.e., 45 deg inclination of the cutting edge).

Face Milling

Face-milling tests were made with the SAE 1020 steel to determine the effect of rake angle on tool life. Fig. 1 shows the setup used in these tests. A 10-in-diam × $2^1/_2$-in-thick cast-iron body with a $^5/_8$-in-square hole was used in conjunction with 18-4-1 HSS tool bits (shown in Fig. 2), to make a single-tooth, fly-type cutter. Only one tooth was used so that tool-life results could

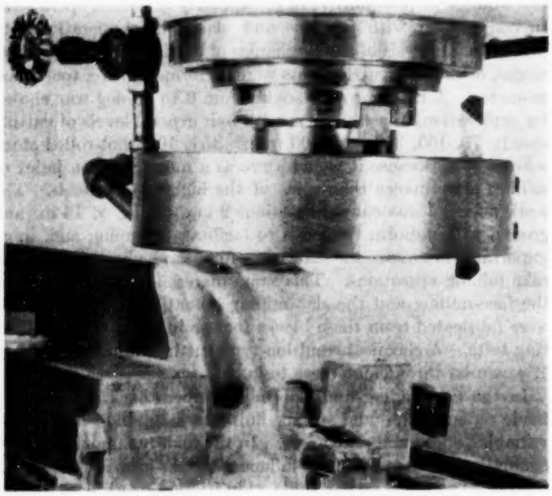

Fig. 1 Experimental Face-Milling Setup for Tool-Life Tests

be obtained with the least amount of stock and time. The tool geometry, for all face-milling tests, was a 45-deg corner angle, 0 deg inclination, and 5 deg clearance. The axial and radial rakes were ground such that 0, 5, 10, 15, 20, 25, 30, 35, 40, and 50 deg true rakes resulted. The face-milling tests were made with 0.009 in. feed per tooth and $^3/_{16}$ in. depth of cut. Tool life was measured in terms of the volume of metal removed to produce a wear land on the tooth flank of 0.020 in. All tools were ground with a 60-grit wheel, lightly stoned by hand, and the cutting edge inspected with a microscope. Owing to the comparatively fragile form of the higher-rake-angle tools, considerable caution was necessary to avoid burning the tools during resharpening.

The curves in Fig. 3 are a generalized or averaged plot of the observed tool life as a function of cutting speed for rakes in increments of 10 deg. They serve to show the over-all trend of the tool-life values, and do not follow the experimental points in exact detail. It can be seen that with a 0-deg rake or a conventional 10-deg-rake cutter, tool life decreases regularly with increasing cutting speed in the conventional manner. However, when larger rake angles are used, tool life at the high cutting speeds is improved while the life at low cutting speed suffers. An investigation of the reason for this break in the tool-life lines was

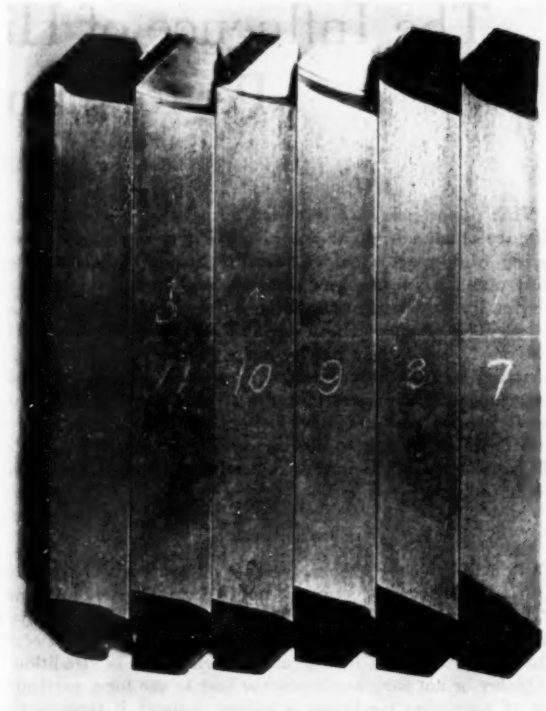

Fig. 2 Typical Tools Used in Face Milling Tool-Life Tests

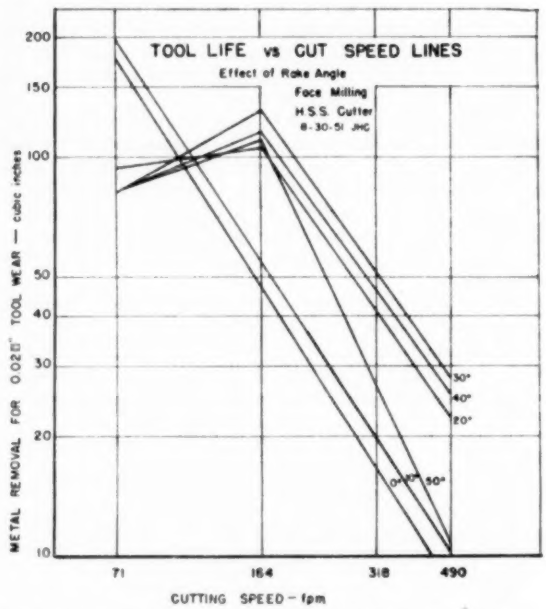

Fig. 3 Effect of Cutting Speed on Tool-Life for Face Milling With Different Rake Angles—Generalized Relationships

made. The size of the built-up edge remaining on the chip was measured, and a visual inspection of the chip and transient surface on the workpiece also made. It was found that the size of

the built-up edge decreased with an increase in speed or rake. Thus with the large built-up edge existing at low cutting speed, the fragile tooth form of the high-rake-angle cutters makes them fail rapidly (by chipping) under the action of this built-up edge and the fragments of it which pass off with the chip and work surface. This accounts for the break in the tool-life lines obtained with these high-rake-angle cutters as seen in Fig. 3. The stronger tooth form of the 0-deg and 10-deg-rake-angle cutters makes them able to stand up without chipping at low cutting speed despite the built-up edge associated with low speed and low rake.

Figs. 4 and 5 show the chips removed in the face-milling operation for the lowest and highest speeds. Each chip consists of the same volume of metal, so it is strikingly evident that the deformation (and thus the force on the tooth) is lessened by an increase in rake angle.

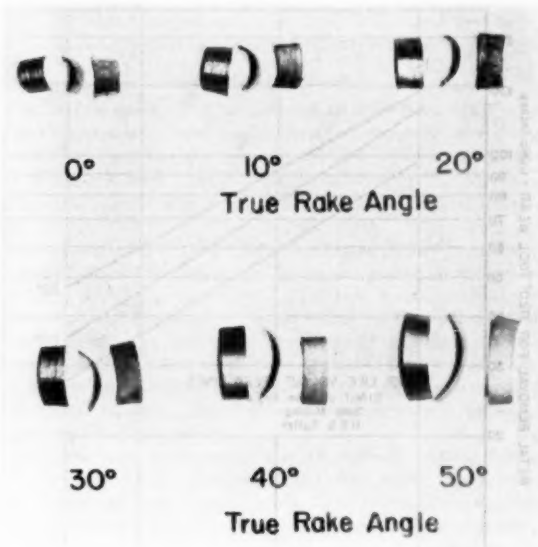

Fig. 4 Chips Resulting From Face-Milling Operation on SAE 1020 Steel at 0.009 Ipr Feed and 71 Fpm Cutting Speed

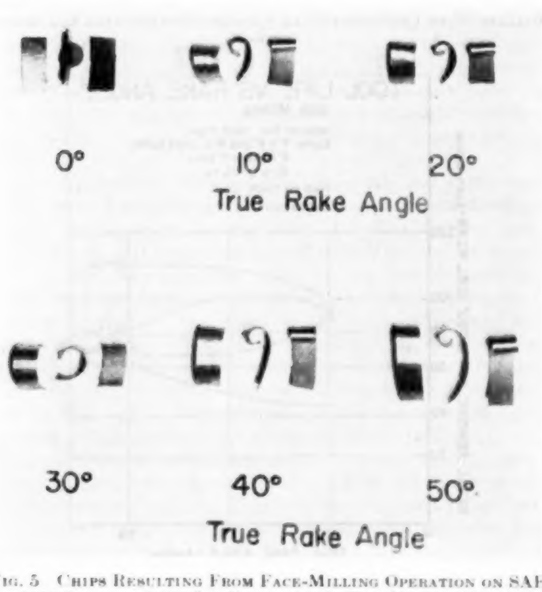

Fig. 5 Chips Resulting From Face-Milling Operation on SAE 1020 Steel at 0.009 Ipr Feed and 490 Fpm Cutting Speed

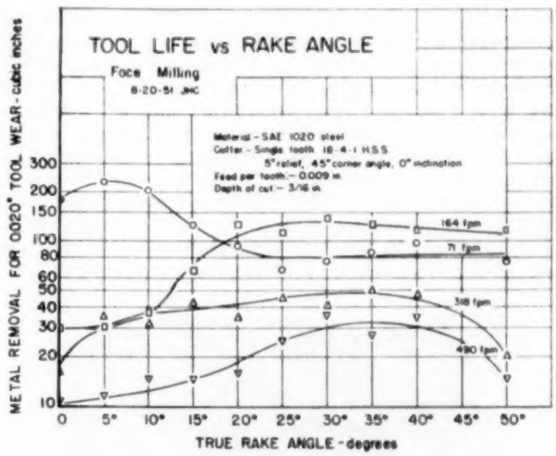

Fig. 6 Effect of Rake Angle on Tool Life for Face Milling—Detailed Relationships

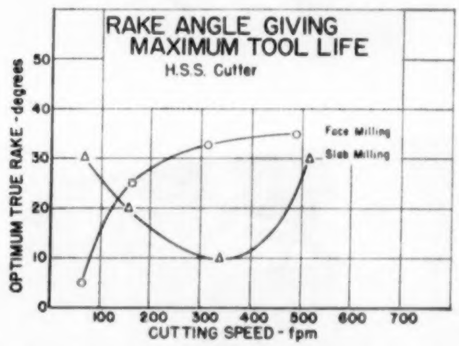

Fig. 7 Effect of Cutting Speed on Optimum Rake Angle for Face and Slab-Milling Operations

Fig. 6 shows the tool-life data plotted as a function of rake angle and presents the detailed trend of the experimental points. It can be seen that with a conventional 10 to 15-deg true rake, best life for these conditions is obtained at low cutting speed, here 71 fpm. Yet if a higher production rate is desired, requiring the use of higher cutting speeds, then as can be seen from the graph, with the conventional 15-deg true rake, tool life will be very poor. However, if at these higher speeds, the rake is increased to about 30 to 40 deg, tool life will be greatly improved. In fact it can be seen that by going to this higher rake angle, as good a cutter life can be obtained at 164 fpm as is obtained at 71 fpm with a conventional 15-deg rake, yet the production rate at the same feed per tooth is more than doubled. It can be seen further that the optimum true rake (the rake angle giving maximum tool life at a given cutting speed) increases with increasing cutting speed. Fig. 7 shows a plot of these values of optimum true rake as a function of cutting speed. The values of optimum true rake for slab milling are also shown and are discussed in the next portion of the paper.

Slab Milling

Slab-milling tests were made to determine the effect of rake

angle on tool life and surface finish. A Cincinnati No. 4 dual power milling machine was used throughout these slab-milling tests. The $5^1/_2$-in. × 8-in. × $1^1/_2$-in., 30-deg-rake, 45-deg-helix slab mill used, Fig. 8, was supplied through the courtesy of Dr. W. R. Frazer and Union Twist Drill Company. For better control of cutter angles and cutter-life measurements, and to keep to a minimum the amount of stock and time required for tool-life testing, only one tooth of the eight on the cutter was used. Seven teeth were ground 0.011 in. below the cutting tooth so that only the one would actually engage the work. The three rakes used, 10, 20, and 30 deg, were all put on the one cutting tooth at successive positions along the cutting edge by grinding the rake face. This made it possible to cut with the three rakes simultaneously on the same cutter as shown in Fig. 9. The clearance angle ground on the slab-milling tooth was 5 deg, as in the face-milling operation. The slab-milling tests were made with 0.009 in. feed per tooth and $^1/_8$-in. depth of cut, using down-milling. Tool life was measured in terms of the volume of metal removed to produce a wear land on the tool flank of 0.020 in. The cutting tooth was ground with a 60-grit wheel, lightly stoned by hand and inspected with a microscope.

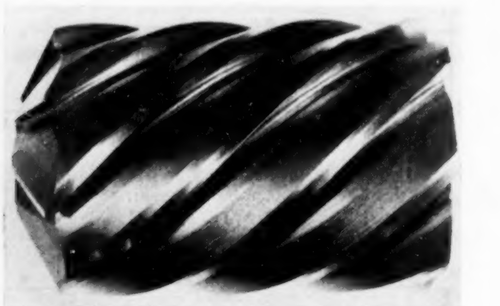

Fig. 8 Cutter Used in Slab-Milling Tool-Life Tests

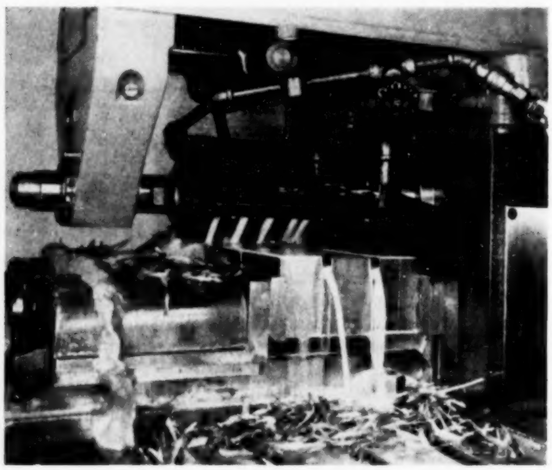

Fig. 9 Experimental Slab-Milling Setup for Tool-Life Tests

The curves in Fig. 10 are a generalized or averaged plot of the observed tool life as a function of cutting speed for the rakes used. Again, they do not follow the experimental points in exact detail. It can be seen that, in general, the higher rake angles give better tool life. Here, in contrast to the face-milling operation, no break in the tool-life lines is evident. However, it was found that no appreciable built-up edge was present initially, even at the low cutting speed, except in the case of the 10-deg rake. The cutting geometry and resulting chip formation are thus evidently more favorable than for face milling.

Fig. 11 shows the same data plotted as a function of rake angle and presents the detailed trend of the experimental points. As an example of the benefit of the higher rakes, it can be seen that at a cutting speed of 68 fpm and a rake of 10 deg the metal removal is 100 cu in. per tooth per grind. Increasing the rake to 20 deg and at the same time increasing the cutting speed to 152 fpm results in the same metal removal per grind, while the production rate is increased 110 per cent. The peculiar drop in

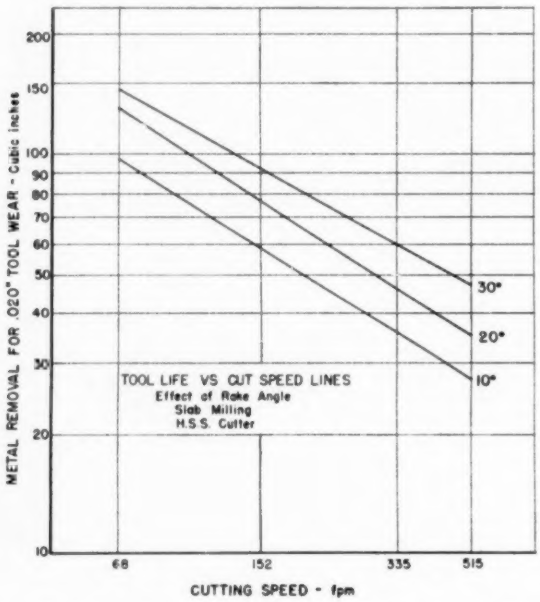

Fig. 10 Effect of Cutting Speed on Tool-Life for Slab Milling With Different Rake Angles—Generalized Relationships

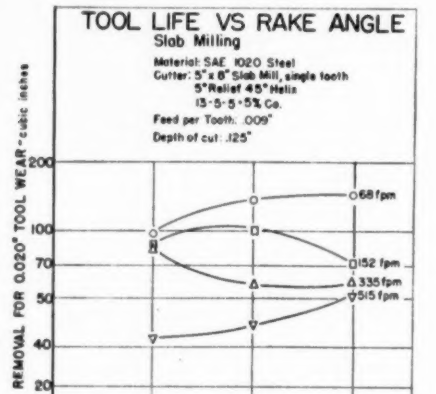

Fig. 11 Effect of Rake Angle on Tool Life for Slab Milling —Detailed Relationships

tool life observed when increasing the rake angle from 20 to 30 deg at a speed of 152 fpm was questioned, and therefore repeat runs were made to check this and other peculiarities in the tool-life relationships. However, the same results were obtained.

It can be seen from Fig. 11 that the effect of cutting speed on the value of optimum rake was rather different for slab milling from that for face milling. The actual relationship is plotted in Fig. 7 for comparison with the face-milling results. The differences may be attributed to the different geometry and smaller built-up edge in slab milling.

Profilometer measurements on the work surface showed that the 30-deg rake at 70 fpm gave the best surface finish while the 20-deg rake was next best and the 10-deg rake produced the worst finish. At higher cutting speeds, surface finish was found to be essentially independent of rake and uniformly good.

Analytical Studies

Tool Dynamometer. Basic investigations of the influence of rake on performance were conducted with a tool dynamometer using an orthogonal cutting arrangement. Tubes with a wall thickness of 0.250 in. were fabricated from the stock used in the slab- and face-milling operations and a chemical emulsion-type coolant used.

The experimental method used in the present study to measure the basic factors influencing tool forces and power consumption has been described elsewhere.[7] The two force components F_c and F_t were measured by means of a two-component tool dynamometer. The additional measurement necessary to calculate the required basic quantities was that of the shear angle ϕ. This shear angle was determined from cutting-ratio measurements in the usual manner. The results of these tests are shown in Table 1 and Figs. 12 and 13.

It may be seen, as shown in Fig. 12, that the cutting forces decrease steadily with increasing rake, due mainly to a large increase in shear angle as seen in Fig. 13. Although the cutting forces decrease and the shear angle increases with increasing rake, the coefficient of friction between chip and tool also increases. Furthermore, it may be seen that the friction coefficient increases quite rapidly with rake above about 40 deg and that the tool forces become nearly independent of rake above about 40 deg. These facts begin to cast some light on the finding that the optimum values of rake, from the point of view of tool life, were found to be in the neighborhood of 30 to 40 deg in the face-milling tests. (In the slab-milling tests the geometry of cutting is quite different from that in the orthogonal cutting tests with the tool dynamometer, making comparisons with the slab-milling data questionable.)

Tool-Chip Interface Temperatures. An analytical study was made of tool-chip interface temperatures using the data obtained in the investigation on the mechanics of cutting, together with measurements of the apparent area of contact between chip and tool, in order to explore the effect of interface temperature on tool life for the range of rake angles used. The interface-temperature calculations were based on the method developed by Trigger and Chao.[8] Since, if an appreciable built-up edge is present their equations do not hold, calculations were made from data obtained at a cutting speed of 272 fpm, where little or no built-up edge exists. The calculated values for interface temperature, plotted against rake angle, are shown in Fig. 14. It may be seen that interface temperature decreases steadily with increasing rake up to a rake angle of about 40 deg, above which point it increases rapidly. This minimum in the temperature curve in the vicinity of 40-deg rake compares favorably with the

[7] "New Methods of Analysis of Machining Processes," by M. E. Merchant and N. Zlatin, *Experimental Stress Analysis*, vol. 3, 1946, pp. 4–27.

TABLE 1 VALUES OF BASIC MECHANICAL QUANTITIES FOR ORTHOGONAL CUTTING USING VARIOUS RAKE ANGLES

Work material: SAE 1020 Width of cut: 0.25 in.
Tool material: 18-4-1 HSS Feed per rev: 0.0063 in.

Observed				Calculated						
V_c, fpm	α, deg	F_c, lb	F_t, lb	r_c	ϕ, deg	ϵ	μ	S_s, psi $(\times 10^{-3})$	E, cu in/hp-min	W_c, in-lb/cu in. $(\times 10^{-3})$
63	0	795	445	0.31	17	3.5	0.56	117	0.80	490
	10	690	400	0.29	17	3.4	0.84	100	0.90	430
	20	590	255	0.30	18	3.1	0.94	94	1.05	360
	30	490	190	0.32	19	2.8	1.22	82	1.25	310
	40	360	75	0.41	23	2.0	1.27	75	1.70	230
	50	310	60	0.51	28	1.5	1.44	73	2.00	200
166	0	930	730	0.16	9	6.2	0.79	80	0.70	580
	10	705	450	0.22	13	4.5	0.91	80	0.90	430
	20	545	285	0.30	17	3.1	1.10	82	1.15	340
	30	435	150	0.36	22	2.3	1.14	81	1.40	280
	40	365	80	0.45	26	1.8	1.30	83	1.70	230
	50	330	80	0.48	27	1.6	2.01	73	1.90	210
272	0	885	775	0.18	10	5.6	0.88	84	0.73	540
	10	670	420	0.23	13	4.3	0.90	80	0.95	420
	20	550	290	0.28	16	3.3	1.09	79	1.15	340
	30	460	150	0.32	18	2.8	1.12	76	1.35	290
	40	365	60	0.39	22	2.1	1.37	76	1.70	230
	50	340	70	0.51	28	1.5	1.85	82	1.80	220
523	0	825	580	0.18	10	5.6	0.75	81	0.79	500
	10	620	345	0.26	15	3.7	0.85	84	1.05	380
	20	535	265	0.29	17	3.2	1.04	79	1.20	330
	30	430	120	0.36	22	2.3	1.19	84	1.45	270
	40	415	180	0.43	25	1.9	1.61	80	1.50	260
	50	390	155	0.49	27	1.5	2.28	80	1.60	240

maxima in the tool-life curves in Fig. 6 for the higher cutting speeds where the built-up edge is small or absent. Thus further basic reasons for the existence of these maxima are evident.

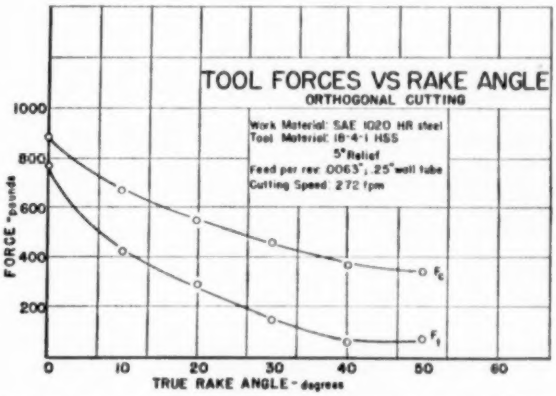

Fig. 12 Effect of Rake Angle on Cutting Forces—Orthogonal Cutting

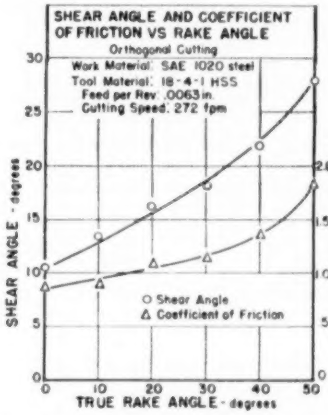

Fig. 13 Effect of Rake Angle on Friction and Shear Angle—Orthogonal Cutting

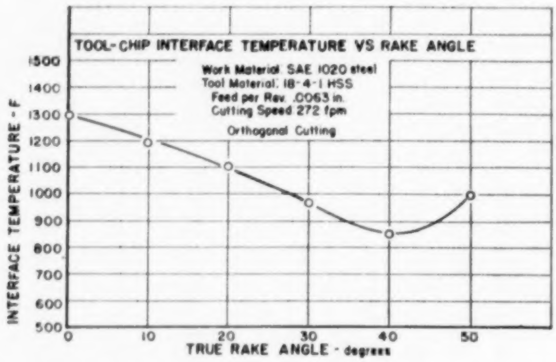

Fig. 14 Effect of Rake Angle on Calculated Interface Temperatures—Orthogonal Cutting

The calculated interface temperature is the sum of two elements—the temperature rise caused by deformation on the shear plane and the temperature rise due to rubbing of the chip on the tool. It was found in the investigations conducted here that while the temperature rise due to rubbing increases with increasing rake angle, the temperature rise due to shear drops more rapidly at first, causing a drop in the interface-temperature curve until the rake angle reaches a value of 40 deg, above which the temperature rise due to rubbing predominates and the interface temperature consequently increases.

It should be pointed out that, since the interface-temperature values in Fig. 14 are calculated values, they are not necessarily exact as to absolute magnitude. However, the relative values, and thus the shape of the curve, can be depended on quite safely. Furthermore, although these values were obtained from orthogonal cutting tests with a tool dynamometer, the directly comparable geometry of chip formation in the face-milling operations, with the 0-deg inclination of the cutting edge and the wide workpiece used, makes them directly applicable to the operation as relative values. In fact, when taking the 45-deg corner angle into consideration, a 0.009-in. table feed per revolution for face-milling with a $^3/_{16}$-in. depth of cut results in an actual undeformed chip thickness of 0.0063 in. and width of 0.265 in. which very closely approximates the conditions used with the tool dynamometer, which were 0.0063 in. feed per revolution and 0.250 in. width of cut. However, the interface-temperature values, as obtained from the tool-dynamometer tests, are not directly applicable to the slab-milling operation, with its quite different geometry of cutting.

Conclusions

From the tool-life tests and the chip-formation and tool-dynamometer studies, it may be concluded that the superior performance (improved tool life and surface finish) obtained with higher rakes and cutting speeds, within the limitations of these tests, is due mainly to the following basic factors:

1 Higher shear angle and correspondingly shorter path of shear.

2 Lessened cutting forces.

3 Lower value of chip-tool interface temperature.

4 Greater metal-removing efficiency and a correspondingly lower value for work done in cutting.

5 Less built-up edge, in turn causing less wear due to abrasion and chipping.

The fact that at higher cutting speeds a maximum tool life is obtained in face-milling with rake angles of 30 to 40 deg is due mainly to the following factors:

1 With very low rake angles, shear angle is low, tooth forces are high, a built-up edge is present, and tool-chip interface temperatures are high, though falling with increasing rake angle.

2 With very high rake angles, the tooth form is fragile, tooth forces are no longer decreasing appreciably with increasing rake, friction coefficient is high and rising rapidly, and tool-chip interface temperatures are high and rising rapidly with increasing rake angle.

Some of the resulting advantages from the use of high-rake-angle cutters are as follows:

1 A higher cutting speed for the same tool life is possible, or—

2 Longer tool life at the same cutting speed (above a certain minimum value of cutting speed) is possible.

3 Better workpiece finish is obtained owing to the absence of built-up edge.

4 Less power is required for the same metal-removal rate.

5 Cutting forces are lower, and consequently there is less deflection and greater accuracy.

Some inherent disadvantages, however, are as follows:

1 More care is required in handling and storage of the cutters because of their comparatively fragile tooth form.

2 Greater care must be exercised in sharpening high-rake cutters as there is less metal present in the tooth to dissipate the heat generated in grinding.

3 High-rake cutters do not stand up well under abrasion so usually cannot be used for roughing or finishing cuts into scale or surfaces with abrasive inclusions. A previously machined or ground surface is best.

4 Cutting fluid used must be generous to aid in heat dissipation.

5 While the use of higher rake gives improved performance when milling SAE 1020 steel and even, as reported by others,[3,4] when milling unhardened high-speed steel, it is understood that with some few types of steel improved performance is not obtained in practice. Further research is needed to clarify this point.

Acknowledgments

The authors wish to express their thanks to Mr. Hans Ernst, Research Director, for his helpful suggestions throughout the course of this research. Their appreciation is also expressed to Mr. E. J. Krabacher, Research Engineer, for his assistance during the course of the chip-tool interface-temperature calculations. The authors also wish to thank Union Twist Drill Company, and Dr. W. R. Frazer of that company, for their kindness in the loan of the milling cutter, without which the slab-milling portion of this research would not have been possible. They also thank Mrs. Margaret Huenefeld and Miss Sue Hannaford for their help in the preparation of the manuscript and illustrations.

Vibrations in Knee-Type Horizontal Milling Machines

By IVAR BENDIXEN,[1] COPENHAGEN, DENMARK

This paper outlines a series of tests on vibrations in knee-type horizontal milling machines, especially in the knee system. The amplitude of the vibrations was determined by means of a capacitive pickup. Systematic investigations on the importance of clamping and unclamping the various elements are made. The influence of damping in the case of resonance is pointed out. An explanation of the vibration accompanying entrance and exit of the milling cutter is indicated. A theory for the presence of low natural frequencies in the knee system is developed. Up- and down-milling are compared.

Introduction

Vibrations in machine tools play an important role. However, very little experimental or theoretical work has been published on the subject and only lathes and lathe tools have been investigated, for instance, by Doi in Japan (1, 2, 3),[2] and Arnold and Chisholm in England (4). After the introduction of welded machine-tool frames a comparison of static rigidity and damping properties between cast construction and welded design was made (5, 6, 7). Eisele conducted some dynamic tests of milling with a three-component measuring instrument, and has made a remarkably good determination of the component cutting forces, but no real determination of the vibratory amplitude (8). A theoretical treatment of the milling process is given by W. Klein who has made a Fourier analysis based on a linear variation of the cutting pressure with the chip thickness (9).

Experimental Equipment

An older Wanderer universal horizontal milling machine and a newer Biernatzki plain horizontal milling machine both having belt drives, the latter with a tightening roller, were used in the tests. Data on these machines are found in Table 1. Three plain milling cutters, as described in Table 2, were used.

The steel workpiece 2.5 × 2.5 × 29.5 in., Fig. 1, was of the following analysis: 0.3 per cent C, 0.32 Si, 0.66 Mn, 0.03 P, and 0.039 S; 150 Bhn, Meyer exponent 2.15.

The amplitude of the vertical vibrations of the milling table was determined by means of a capacitive pickup built on the seismograph principle and connected to a Philips cathode-ray oscillograph GM-3156.

Experimental Procedure

The workpiece was fastened to the milling table by means of two clamping bolts and remained in the same position during the tests. The space between the bolts was divided into 26 equal parts and the different cutter positions indicated by numbers from 0 to 26, Fig. 1. The pickup was clamped to one end of the milling table and remained in this position throughout the tests.

All tests were run dry under the following machine conditions:

I All elements clamped.
II All elements clamped but knee unclamped from column.
III All elements clamped but saddle unclamped from knee.
IV All elements clamped but overarm unclamped from column.
V All elements clamped and overarm brace added.

Discussion of Tests

The relation of the vibrations to the clamping of the elements and to the position of the table in lengthwise and transverse directions was investigated first. Tests were made with:

1 Cutter A operating under machine conditions I, II, III, and IV.
2 Cutter B operating under machine conditions I, II, and V.
3 Cutter C operating under machine conditions I, II, and III.

The plotted curves are characteristic examples from about 800 tests (10). The curves may be divided into two groups, namely, those having a steady falling tendency and those having a maximum in the middle.

The following average values for all tests were found:

1 When machining with cutter B (3 in. diam, 14 teeth, 3 in. width) the amplitude was increased: 20–24 per cent upon changing the machine condition from V to I, i.e., removing the overarm

TABLE 1 MILLING-MACHINE DATA

		Wanderer milling machine	Biernatzki milling machine
Table size		47 in. × 11.25 in.	39.4 in. × 11.8 in.
Feed range	Longitudinal, in.	31.5	25.6
	Cross, in.	10.8	8.3
	Vertical, in.	17.75	15
Table feeds	Number of changes	10	8
	Range, ipm	0.31 to 5.9	0.4 to 6.5
Spindle speeds	Number of changes	12	6
	Range, rpm	16–360	50–800
Type of guides	Table	Prismatic guides (50°)	Prismatic guides (45°)
	Saddle	Prismatic guides (50°)	Flat bed way 1.1 in. thickness
	Knee	Prismatic guides (50°)	Flat bed way 0.95 in. thickness

TABLE 2 MILLING-CUTTER DATA

	Milling cutter A	Milling cutter B	Milling cutter C	Milling cutter D
Number of teeth	6	14	14	1
Diameter of cutter, in.	3	3	3.5	5.5
Width of cutter, in.	3	3	1.4	0.59
Diameter of arbor, in.	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{1}{8}$
Helix angle, deg	45	10	15	0
True rake angle, deg	+15	+2	0	−5, −2, +3
Clearance angle	10	4	2	4
Material of cutter	HSS Rockwell C63	Carbon tool steel Rockwell C64	HSS Rockwell C63	Carbide steel cutting grade

[1] Mechanical Engineer, Machine Tool Laboratory, Technical University.

[2] Numbers in parentheses refer to Bibliography at end of paper.

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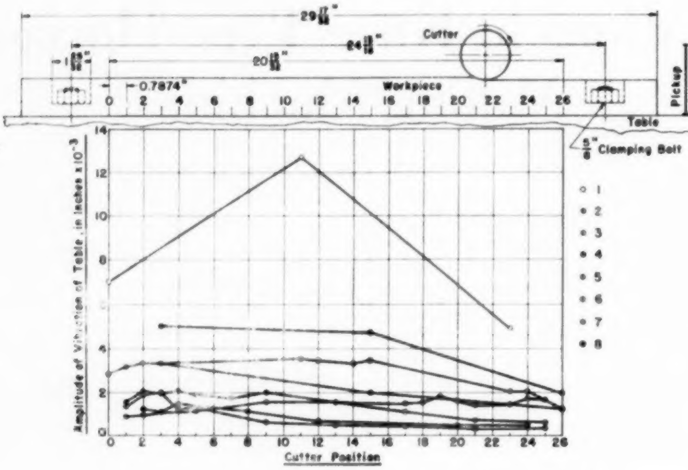


Curve	Milling cutter	Machine condition	Feed per Tooth, in.	Depth of cut, in.	F, in.	E, in.
1	A	I	0.004	0.080	15.6	19.3
2	A	II	0.004	0.080	15.6	19.3
3	A	I	0.004	0.080	5.5	9.25
4	A	II	0.004	0.080	5.5	9.25
5	B	II	0.002	0.060	10.25	21.8
6	B	V	0.002	0.060	16.15	21.8
7	B	II	0.002	0.060	16.15	21.8
8	C	III	0.002	0.040	5.7	

Fig. 1 Relationship Between Amplitude of Table Vibration and Position of Milling Cutter—Wanderer Machine
(See Fig. 2 for dimensions F and E and Table 2 for cutter data.)

brace; and 60–65 per cent upon changing the machine condition from I to II, i.e., unclamping the knee from the column.

2 During the tests with cutter C (3.5 in. diam, 14 teeth, 1.4 in. width) the amplitude was increased: 100–135 per cent when changing the machine condition from I to II; and 10–20 per cent by changing the machine condition from I to III, i.e., unclamping the saddle from the knee.

3 Tests with cutter A (3 in. diam, 6 teeth, 3 in. width) registered an increase in the amplitude of 10–15 per cent when changing the machine condition from I to IV, i.e., unclamping the overarm from the column, but indicated a decrease to 60 per cent when changing the machine condition from I to II, (4–5 per cent in average), and 2–3 per cent decrease when changing the machine condition from I to III.

This means that in some cases the most "hingy" machine gives the greatest amplitudes and in other cases the smallest amplitudes. That a statically less rigid column can be better than a more rigid one with regard to vibrations has also been pointed out by Kettner (5). Throughout the tests no violent vibrations or rough surfaces were registered when milling between the index marks on the workpiece. Yet some tests with cutter A running at 72 rpm gave very heavy vibrations when milling took place just over the clamping bolts holding the workpiece, i.e., outside the index marks, Fig. 1. These vibrations were minimized when the elements of the machine were unclamped, which action was necessary in order to avoid destruction of the cutter over these sections.

To investigate if resonance might exist in some tests, an experimental determination of the natural frequencies of the knee system was made. For this purpose an electric motor which was provided with an unbalance, was clamped to the milling-machine table at the place where otherwise the milling cutter worked. The motor could produce considerable impulse

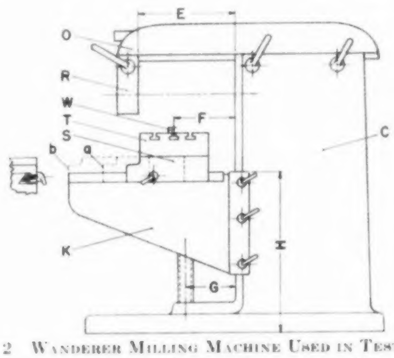


Fig. 2 Wanderer Milling Machine Used in Tests
(F = distance between column and center of cutter; E = distance between column and arbor support; O = overarm; R = arbor support; W = workpiece; T = table; S = saddle; K = knee; C = column; G = 10.8 in.; H = 40.5 in. a and b are saddle position limits for one series of tests. Details of saddle-clamping arrangement are shown at left.)

but was rather light so that the weight itself should have little influence on the results. The milling machine would vibrate in resonance under impulses from this motor at frequencies equal to the natural frequencies of the machine. The motor speed was varied continuously between 0 and 3000 rpm. When the saddle was positioned between a and b, Fig. 2, a sharp resonance section around 1075 rpm was noticed. Another section around 1800 rpm and a somewhat wider range around 2200 rpm, Fig. 3, was found, the latter being greater in amplitude the nearer the saddle was positioned to b for machine conditions I, III, IV, and V. For machine condition II, the knee unclamped from the column, the resonance amplitudes were smaller, and the sections displaced a little toward lower rpm. When the saddle was close to the column, Fig. 4, for machine conditions

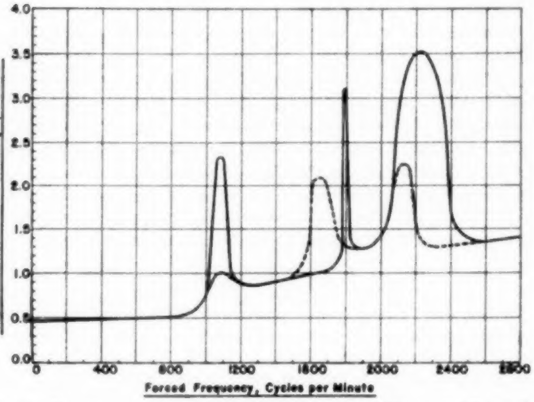


FIG. 3 RESONANCE CURVES FOR VIBRATION OF MILLING-MACHINE TABLE

(Saddle positioned between a and b. Dotted line, Machine condition II; solid line, Machine condition I, III, IV, V.)

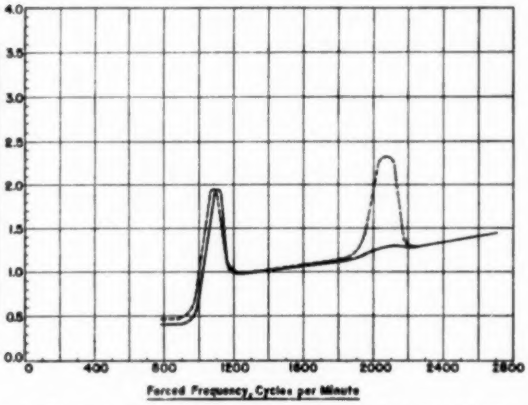


FIG. 4 RESONANCE CURVES FOR VIBRATION OF MILLING-MACHINE TABLE

(Saddle positioned close to column. Dotted line, Machine condition II; solid line, Machine condition I, III, IV, V.)

I, III, IV, and V, a resonance section occurred around 1100 rpm, while the section around 2100 rpm nearly disappeared. Machine condition II gave resonance between 1050–1100 rpm and 2000–2100 rpm. Around 1800 rpm nothing was to be noticed in any case. A movement of the table in the lengthwise direction affected only the amplitudes and not the position of the resonance sections. If the knee were lowered, the resonance sections would be displaced slightly toward higher rpm. The position of the resonance sections was unaffected by running the milling machine idle or even by feeding the table simultaneously.

The fundamental forced frequency in these milling tests was found to be the product of the number of teeth multiplied by the rpm of the cutter. In the case when one of the natural frequencies is a multiple of this forced frequency, resonance will occur. The forced frequencies multiplied by the smallest figure that gives a product in the vicinity of a natural frequency are listed in Table 3. In the table the cases giving a steadily falling curve of the amplitudes of vibration are marked p, and the cases giving a maximum, when milling in the middle of the work, are marked q. The tabulation shows that in case of resonance the curve most frequently has a maximum in the middle; otherwise it is steadily falling.

This difference in the shape of the curves could be caused by damping. Practically, damping will affect the amplitude of the vibrations only in the case of resonance. For a single mass we have the following expression for the vibratory amplitude

$$X = \frac{\frac{P}{k}}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_m}\right)^2\right]^2 + \rho_t^2\left(\frac{\omega}{\omega_m}\right)^2}}$$

where P is the exciting force, k the rigidity or spring scale, ρ_t the total relative damping, ω the forced frequency, and ω_m the natural frequency. The relative damping is connected with the logarithmic decrement δ by the equation

$$\rho = \frac{\delta}{\pi}$$

In our case ρ_t is of the order of magnitude 0.05. The foregoing expression is illustrated in Fig. 5.

At speeds below the critical speed in the "static range" the vibration amplitude is determined mainly by the rigidity of

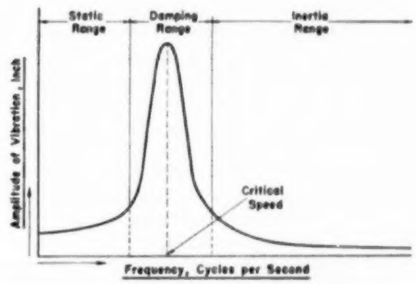


FIG. 5 FUNDAMENTAL RELATIONSHIP BETWEEN VIBRATORY AMPLITUDE AND FREQUENCY

TABLE 3

Milling cutter A: RPM of cutter × no. of teeth × (5 or 3)	Milling cutter A: Machine condition II; saddle between a and b	Milling cutter A: Machine condition I, III, IV, V; saddle between a and b	Milling cutters B and C: Rpm of cutter × no. of teeth × (2 or 1)	Milling cutters B and C: Machine condition II; saddle between a and b	Milling cutters B and C: Machine condition I, III, IV, V; saddle close to column
69 × 6 × 5 = 2070	p, r	q, r	59 × 14 × 2 = 1652		p
71 × 6 × 5 = 2130	p, r	q, r	78 × 14 × 2 = 2184		p
72 × 6 × 5 = 2160	q, r	q, r	110 × 14 × 1 = 1540	q, r	p
73 × 6 × 5 = 2190	p	q, r			
74 × 6 × 5 = 2220	p	q, r			
100 × 6 × 3 = 1800	p	q, r			
125 × 6 × 3 = 2250	p	q, r			
132 × 6 × 3 = 2376	p	q, r			

NOTES:

p, Amplitude of vibrations grew evenly with increasing distance between milling cutter and pickup.
q, Maximum vibrations occurred when milling took place near center of workpiece.
r, Natural frequency equaled multiple of product of number of cutter teeth and rpm of milling cutter.

the system. In other words, with an increase in stiffness of the milling machine there is a decrease in the amplitude of vibration. In the "damping range" around the critical speed the amplitude is determined mainly by the various sources of damping within the machine. Above the critical speed in the "inertia range" the amplitude is determined by the masses of the table and the other elements.

The total damping effect derives mainly from two sources (11, 12):

1 Damping due to friction between the sliding surfaces.

2 Frictional damping which may be affected to a considerable degree by the small relative motions made between elements clamped to each other.

Friction is then of deciding influence for dampening these vibrations (4, 14).

Of the components given, damping due to friction in the guides of the table may explain the maximum attained in the middle of the curves. When milling was done near the center of the workpiece, the table was simultaneously in its central position, where the guides of the old Wanderer machine were worn the most. Consequently, the friction and therefore the damping were least at this point. When milling took place just over the clamping bolts, very heavy vibrations occurred at resonant frequencies, since frictional damping due to relative motion between the workpiece and table disappeared in the immediate vicinity of the clamping bolts. Vibrations caused by the milling cutter entering and leaving the workpiece have often been noticed, for instance, by Eisele (8).

From Fig. 1, curves 1, 2, 3, and 4, where the relative position of the arbor support with respect to the cutter was maintained with the arbor support just to the outside of the cutter, it will be seen that a greater space between column and table, of necessity accompanied by location of the cutter further out on the arbor, means a greater amplitude. During the tests plotted in Fig. 1, curves 5, 6, and 7, the arbor support remained fixed in its outer position, while the cutter was situated at various locations on the arbor. The greatest amplitudes occurred when the cutter was located in the middle of the arbor, although the work here is situated directly above the knee elevating screw.

These tests could be summarized as follows: The knee system in an ordinary knee-type milling machine has a certain number of natural frequencies in the range 0 to 3000 cycles per minute (cpm) which might give resonance. This resonance will not necessarily produce serious vibrations while milling the workpiece in the region between the clamping bolts, but when the cutter is passing directly over the clamping bolts, greater amplitudes of vibration will occur. A substantial part of the damping is due to the friction in the guides of the table. When cutting occurs outside the regions of resonant frequencies, the amplitudes will be smaller the more securely the various elements are connected or clamped to each other. While the knee was secured to the column by means of three clamping levers pressing against a gib, the saddle was held by only one, Fig. 2. The tests indicated that the single saddle-clamping lever did not permit effective clamping. At resonant frequencies under machine conditions I, III, IV, and V the knee may become loosened from the column and under this new condition resonance might not exist because of a shifting of the resonant frequencies to different values. However, resonant-frequency ranges under machine conditions I, III, IV, and V may sometimes be the same as, or may overlap, those under machine condition II so that, even with the knee having become unclamped, the machine may still be operating at a resonant frequency. Nevertheless, the amplitude of vibration for machine condition II is smaller than for machine conditions I, III, IV, and V.

The most effective way of shifting out of a resonance section is to change the rpm of the cutter, or to use a cutter with a different number of teeth. A change in the feed, depth of cut, or width of cut may result in a shift out of a resonance section, since a small change in the rpm may be caused by slippage in the belt drive. For example, milling with cutter B at 0.160 in. depth of cut with a speed of 100 rpm under machine condition II resulted in 4–5 per cent smaller amplitudes than under machine condition I, while at 0.030 in. depth of cut the speed increased to 110 rpm and the amplitudes under machine condition II were 35–40 per cent greater than under machine condition I.

What Is Vibrating?

Of the three cutting-force components the component in the feed direction usually is by far the greatest (8, 13). M. Kurrein has shown how the knee system will be deflected by such a horizontal force (13). The sketches in Fig. 6 (top), show exaggerated play between various machine elements and illustrate typical relative positions induced and sustained by torsional moments caused by the cutting forces. It can be seen from the points of contact that forces must be concentrated quite at the prismatic edges. The calculated deflection for such a loading condition is very large. Fig. 7 illustrates the deflection on a model of the prismatic guides of the column (Wanderer milling

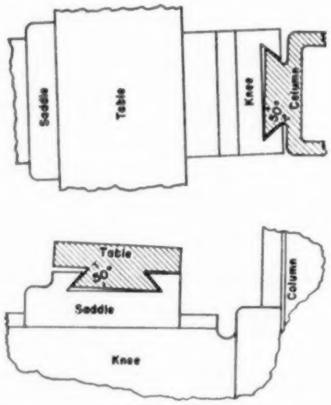


Fig. 6 Exaggerated Sketch Depicting Possible Changes of Position of Knee With Respect to Column and of Table With Respect to Saddle Under Cutting Conditions

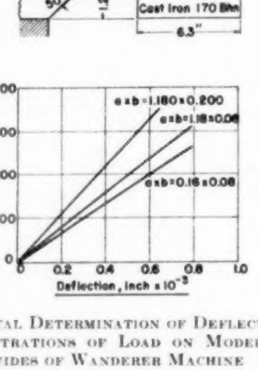
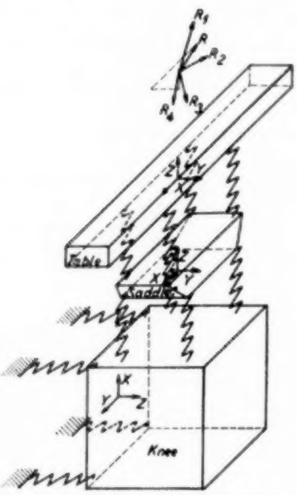


Fig. 7 Experimental Determination of Deflection at A With Different Concentrations of Load on Model of Prismatic Guides of Wanderer Machine

machine). The deflection was found experimentally for various concentrations of force, and is, within limits, independent of the length of the prism. Consequently there is between column and knee, knee and saddle, and saddle and table a number of rather weak "springs." Therefore a vibration model of the knee system will be similar to the sketch shown in Fig. 8. The elements must be guided so that only longitudinal vibrations along the *Z*-axis and torsional vibrations in the *X-Z-* and *Y-Z-*planes are possible. Friction is then of deciding influence for dampening these vibrations (4, 14).

As long as resonance exists in both ends of the work and not only in one, the lead screw, fixed at one end of the table with respect to axial displacement in either direction, is of no importance for natural frequencies within the range 0–3000 cpm. Similarly, the elevating screw exerts no decisive influence. A lowering or raising of the knee did not affect the natural frequencies much.



R_1, up-milling (positive rake angle, great depth of cut and feed per tooth)
R_2, up-milling (negative rake angle)
R_3, down-milling (negative rake angle)
R_4, down-milling (positive rake angle)

FIG. 8 VIBRATION MODEL OF KNEE SYSTEM OF UNIVERSAL MILLING MACHINE WITH PRISMATIC GUIDES

(R, direction of resultant cutting force when milling with a slabmill having spiral teeth—up-milling, positive rake angle, moderate depth of cut and feed per tooth. R_1-R_2-R_3-R_4, direction of resultant cutting forces when milling with a slabmill having straight teeth.)

Curve	Depth of cut, in.	True rake angle, deg
1	0.040	+3
2	0.060	−5
3	0.040	+3
4	0.060	−2

FIG. 9 AMPLITUDE OF VIBRATION OF MILLING MACHINE TABLE IN RELATION TO CUTTING SPEED

(Milling cutter D; machine condition 1; 0.005 in. feed per tooth. Solid line, up-milling; dotted line, down-milling.)

The following points indicate that the "weak" prismatic guides cause the series of low natural frequencies:

1 When the saddle is close to the column, the resonance section around 2100 cpm exists only under machine condition II. The moment arms of the acting forces are so short here that the knee must be unclamped if the machine elements are to assume relative positions such as shown in Fig. 6.

2 In down-milling the vibratory amplitudes are considerably smaller than in up-milling, Fig. 9. This can be explained by the direction of the resulting cutting forces in down-milling which have a direction such that no "upsetting" or relocation of machine elements to relative positions such as shown in Fig. 6, takes place. The springs may then constitute rigid supports.

3 The Biernatzki milling machine, with prismatic guides only in the milling table (Table 1) had only two small resonance sections (less than 1500 cpm) and no resonance section around 2100 cpm.

CONCLUSIONS

1 In the three frequency ranges the smallest amplitudes occur as follows with the:

(*a*) Greatest static rigidity in the static range.
(*b*) Greatest damping in the damping range.
(*c*) Greatest mass in the inertia range.

The static range is of special importance with regard to machine-tool elements.

2 The rigidity of the setup under operating conditions and the direction of the cutting forces which tend to "upset" or relocate the various machine elements to relative positions as shown in Fig. 6, are of basic importance with regard to good milling-machine performance.

3 The friction between the milling-machine elements has the greatest damping effect.

4 The amplitude of vibration decreases rapidly with increasing cutting speed and, at higher speeds, appeared to be unaffected by rake angle, depth of cut, and either up- or down-milling.

ACKNOWLEDGMENT

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Discussion

R. S. Hahn.[3] The author is to be encouraged to continue his good work because there is much to be learned about the vibration of machine tools.

There are several points which might be discussed in this paper. In the author's discussion of tests, the statement is made that under certain conditions "the most hingy machine gives the greatest amplitude and in other cases the smallest amplitude." It might be inferred from this that rigidity is sometimes undesirable in a machine tool from a vibration point of view. It sometimes happens (in a multimode system and may readily be explained) that when the rigidity of some machine element is reduced, greater damping appears at the cutter, and often the increase in damping outweighs the reduction in rigidity, thereby resulting in less vibration. Consequently, a loss in rigidity in itself is always undesirable but only when the system is carefully designed vibrationally to get damping, should any loss in rigidity be tolerated.

In studying the vibration of machine tools the writer has found it essential to determine the natural modes of vibration of the machine in order to locate damping devices properly and to obtain an understanding of the vibration. It would have been desirable if the author had found the mode (shape or configuration) of vibration corresponding to each of the natural frequencies observed. When the modes of vibration are known it is much easier to evaluate the relative rigidities of the machine. The modes may be determined by judiciously placed pickups and by tapping at various locations and in various directions.

As a simple example to illustrate the principle, consider a free-free bar. If a pickup is placed at one of the nodes (at about 0.22 l) it will not pick up the lowest mode but only higher modes. If the bar is tapped at one of the fundamental nodes only higher modes will be excited. By observing the position of tapping and the frequency picked up it is possible to search out the nodes and antinodes of any particular mode of vibration.

F. W. Lucht.[4] During 1945 we made a series of cutting tests to obtain "An Analysis of Flywheel Effect When Face Milling Steel With Carbides." This unpublished work was done on a Kearney & Trecker 5HM vertical milling machine using four different-size flywheels, and six simultaneous oscillographic traces were made covering each of 123 different sets of operating conditions.

Vibrations at one end of the table were included among the original oscillogram readings, but it was soon found that they also included vibrations transmitted through the concrete floor from production machines operating at a distance from the milling machine. This oscillogram trace proved to be confusing and was dropped from the records after the first few runs were made.

It was found by the aid of a portable vibration meter that the optimum location to obtain a vibration reading for this work was to position the strain gage to measure the displacement between the spindle head and the column of the machine approximately in the horizontal plane of the face of the spindle nose.

It was found that vibration had a tremendous effect on cutter life, which was used as a measure of cutter performance, for each one of the operating conditions. The results of these tests clearly indicated the importance of applying suitable flywheel effect to the machine spindle to eliminate detrimental vibration and again obtain optimum milling-machine performance. It may be of interest that several of the basic principles evolved or confirmed from this analysis have been incorporated in the design of the machines as produced by the leading milling-machine manufacturers today.

The author did not make any reference in his paper to difficulty he might have had when he located his vibration pickup at one end of the table. It may be possible his lighter speed, feed, depth of cut combination did not create the same problem we had when using 0.010-in. feed per tooth with one to eight teeth in the cutter and 0.150-in. depth of cut.

K. E. Wetzel.[5] This paper is an interesting attempt to analyze the source of low-frequency vibrations in machine tools and their efficient damping. In connection with the author's investigations, we ran a number of similar tests in the machine-tool laboratory at M.I.T.

To determine the influence of the single elements on the resulting vibrations is a good method to eliminate these elements by unclamping, as long as the static conditions are not changed thereby. If gravity fixes the unclamped elements in their normal position, we had a statically stable system and could expect to get comparable results. However, when the knee was unclamped from the column the conditions were fundamentally changed. Here we had no more static balance, gravity could not fix the elements in their proper position, and the system became unstable. Knee, saddle, and table were then supported only by the elevator screw. The vibration relations were then mainly determined by the direction and amplitude of the dynamic forces, the friction damping disappearing almost completely. As a consequence, vibrations became very large and the excessively loaded prismatic guides could not decrease the amplitudes.

There is no doubt that this case has great importance in machining and it is worth while to direct the attention of designers to this point. If gravity acts in the direction of the guides we should try to maintain some friction damping even when there is wear or clearance in the guide surfaces. The geometrical form of the guiding elements should be determined with this view in mind. For increasing the damping effect in the guides perhaps the design of plastic plates will be useful.

The author's tests were concerned with the low frequencies of elements where inertia forces could be neglected. Such vibrations can be reduced by increasing the rigidity of the machine, which usually means also increasing the mass. Friction damping was found as the most important means for decreasing these vibration amplitudes. It demands large areas of contact over which small motions of the clamped elements against one another can take place.

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The low frequencies which cause resonance in the frame and other heavy elements of the machine are important and dangerous for the machining process, but they are not the only ones. We ran tests with higher forced frequencies and found the condition shown in Fig. 10 of this discussion.

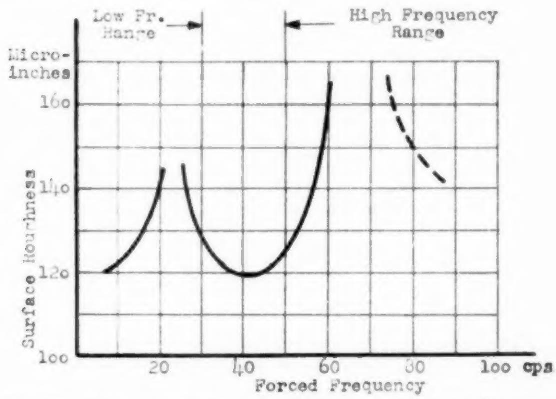
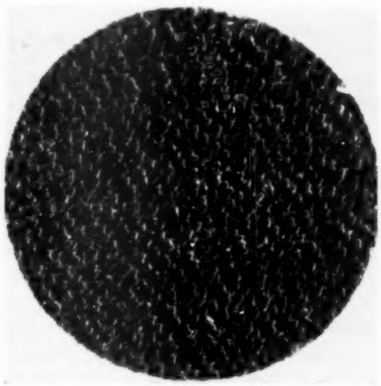


Fig. 10 Vibration Test of a Knee-Type Milling Machine
(Milling conditions: High-speed milling cutter; down milling. Speed, 35–250 rpm; feed, 0.001 in. per tooth; depth, 0.06 in.)



Speed direction

Fig. 11 Chatter in Metal Cutting: Surface Finish
(Machining conditions: Carbide tool. Speed, 42 rpm; surface speed, 60 fpm; diameter, 8 in.; depth, 0.16 in., chatter frequency about 600 cps.)

As a measure of the amplitude we took the surface roughness of our workpiece, which we assumed to be a reliable criterion. The method is not as exact as a vibration pickup would be; for example, we could not calculate the value of our maximum frequency. However, it does give a fair picture of the situation. We can see a resonance in the low-frequency range which also is evident in the author's illustrations; then the roughness decreases rapidly and stays at a low value. By increasing the frequency further the roughness increases again. The dotted line shows the probable manner in which it will decrease; we cannot be quite sure of the high-frequency tests (80 cps) because we could run the machine at these speeds for a very short time only.

This is believed to be a characteristic picture of the vibration situation of a system containing both elements with high masses and slow motions (i.e., table) as well as elements with smaller masses but faster motions (i.e., spindle). Usually the ranges will not be so sharply divided. Frame, knee, saddle, and table, with their low natural frequencies, are responsible for the first maximum. The rotating elements like the gears, wheels, the tool, and the arbor have much higher natural frequencies and run in resonance with forced frequencies in the upper range.

That the frequencies in the metal-cutting process can be quite high is shown clearly in Fig. 11 of this discussion. Here we have a typical example of chatter, a phenomenon well known to every metal-cutting expert. In the case shown, we have a chatter frequency of about 600 cps and it is a fatal error to think that such vibrations will not do any harm because their amplitudes are small. Here we see the effect on the workpiece surface. We cannot see the destruction of the cutting edge, but it is well known that tools, especially carbide ones, are very sensitive to such vibrations. They cause small cracks which destroy the structure of the tool and cause rapid breakdown.

We suppose this chatter arises by a very fast alternating tool deflection owing to the fact that static equilibrium is impossible to maintain with the slight changes in cutting force that take place all the time. Friction of tool and workpiece can give us a similar effect, known under the name of "stick-slip." Here an alternating sticking and slipping causes a forced frequency which is transferred to the cutting edge and occurs on the workpiece as chatter marks. Efficient damping of these vibrations is difficult because the motions are very small. A successful solution for some cases is the inertia-damping proposed by Dr. Hahn.

Troels Warming.[6] The writer was interested in the experimental determination of the natural frequencies of the milling machine. This was done by mounting a variable-speed unbalanced motor on the work table and measuring the amplitudes at the end of the table. It would be a tremendous job to calculate these frequencies and a lot of loose assumptions would have to be made. Unfortunately, the pickup used was very large so it might have changed the system. The pickups available now have about the size of a package of cigarettes. It would have been very interesting to move the pickup around on the milling machine or maybe use several pickups at the same time. That way the magnitude and direction of the amplitudes at several important points could have been determined, and this information is essential before any redesign of the machine can be undertaken. When material is added somewhere, the increased stiffness may raise some natural frequencies while the increased weight may lower other frequencies. If one knows the amplitudes at several points, it may be possible to predict these effects.

The location of the pickup also explains the "steady falling tendency" of most of the curves. It is evident that as the end of the table with the pickup moves out, the measured amplitude will increase even if other and more important amplitudes are unchanged.

It seems reasonable that the damping should be maximum with some medium amount of clamping. When two parts are completely free to move relative to each other, there is no energy absorbed during such motion. If they are clamped so tight that no relative movement is possible there is again no energy absorbed, but if there is a limited amount of friction between the parts a large amount of energy may be absorbed. This is similar to the effect of torsional vibration dampers on crankshafts. A definite value for the damping coefficient is necessary to get minimum vibration in the crankshaft. On milling machines, however, this damping is caused by dry friction so the vibrations will wear out the parts.

The author mentions some very weak springs caused by the

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prismatic edges. He assumes that the load is concentrated at the very tip of these edges. This of course will give an infinitely soft spring, but the resulting deflection will move the load onto the stronger part of the prism. Thus the over-all effect is the same as that of a small increase to the clearance in the guides. It is known that such a clearance will cause a very low spring scale which, however, increases with increasing amplitude.

Author's Closure

The author thanks the discussers for the interest they have shown in the paper. Mr. Wetzel's comparative experiments at the Machine Tool Laboratory of the Massachusetts Institute of Technology are appreciated very much.

For machine tools several dangerous vibration ranges are present: (*a*) Low-frequency range with natural frequencies up to 50 cps (frame, knee system, and other heavy machine elements); (*b*) vibration range with natural frequencies from 50 cps up to a few hundred cycles (rotating elements like gears, wheels, spindle, and tool); (*c*) high-frequency range with natural frequencies from a few hundred cps up to several thousand cycles (vibrations of the walls of the machine elements like membranes and self-excited vibrations of the tool).

Mr. Wetzel shows a nice example of chatter marks (his Fig. 11) caused by self-excited vibrations of the tool. The cause of such vibrations can, as Professor Arnold points out, be found in the decrease of the cutting force with increasing cutting speed. However, speed ranges can be found in which the cutting force increases with the speed, especially in case of free-cutting steel. In such ranges the induction of self-excited vibrations will be counteracted. Another cause of chatter may be found in the occurrence of dry friction between tool and workpiece.

The author agrees fully with Mr. Hahn and Professor Warming in the desirability of determining the mode for each of the natural frequencies. In Germany, for instance, they too, during recent years, have worked with such problems and among other things have found the mode of a radial drilling machine at 35 cps and the impulse working at the lower motor bearing. However, the economics did not allow experiments to be carried to such an extent. On the basis of his experiments the author has shown the vibration model in Fig. 8 from which it should be possible to estimate the modes, or in any case to use it as basic to a later exact determination of corresponding values of natural frequencies and modes.

Not only for frame and knee system will it be important to know the natural frequencies and the modes, but also of the gear system, especially the spindle and the arbor, in which torsional vibrations can occur. It is of great interest to know these conditions.

If the position of the fundamental nodes has been determined on the system spindle arbor, a gear wheel can be placed on the spindle and milling cutter in the fundamental nodes, and demonstrate that impulses on these elements just excite higher and less dangerous modes. This may require placing flywheels on both sides of the gear wheel and tool.

When milling with cemented carbide it is necessary under all conditions to use flywheels. Mr. Lucht asks about the effect of flywheels. The influence of flywheels on the amplitude of vibrations of the milling table in milling with lower cutting speed is disappearing (Fig. 48 in Bibliography 10). During the experiments, Fig. 9 of the paper, it was necessary because of the durability of the cemented carbide to use flywheels. Flywheels have to maintain constant speed of spindle and avoid occurrence of torsional vibrations.

When discussing flywheels on milling machines, the following must be considered:

1 The teeth of the gear wheel are made with a certain error. Let us say that the wheel which meshes with the gear wheel of the spindle is Δ in. too thick. If the contact ratio is m_p, number of teeth N, the pitch diameter d, and the number of revolutions n, each tooth is in mesh along a part of the pitch circle

$$\frac{\pi d}{N} m_p$$

Each tooth is in mesh in

$$\frac{\dfrac{\pi d m_p}{N}}{\pi d n} = \frac{m_p}{Nn} \text{ min}$$

When a tooth which is too thick passes into mesh, it will try to press the wheel a little forward so that it is the only tooth in mesh. This tooth with regard to dynamic can be considered to be loaded with the inertia of the whole gear wheel. If the flywheel effect of the gear system measured on the axle of the gear wheel in question is GD^2, the deformed tooth is possessed with an inertia mass

$$\frac{GD^2}{gd^2}$$

where g = acceleration of gravity.

As the deformed tooth runs along the line of contact, the inertia mass of the spindle system will try to move it Δ in. backwards and hereby affect the tooth with a certain force. If this force is variable after a sine curve, it can be calculated, as the tooth immediately before and after engagement will be in its normal position. The frequency of the tooth vibrations is

$$\omega = \frac{2\pi}{60} \frac{Nn}{m_p}$$

The greatest force

$$P = \frac{GD^2}{gd^2} \Delta \left(\frac{2\pi}{60} \frac{Nn}{m_p}\right)^2$$

It will be seen that P increases greatly with increasing number of revolutions. To gain constant speed of the spindle with the great rpm which are used in milling with cemented carbide, it is necessary to give the spindle great flywheel effect.

2 If the flywheel effect of the spindle system is small, a constant wind-up and resiliency will occur during machining of the elastic system consisting of axles and teeth on gear wheels, resulting in "tooth shocks" and torsional vibrations in the whole gear system. These dangerous effects on the cemented carbide are eliminated by increasing the flywheel effect of the spindle system by means of flywheels which are positioned as near to the tool as possible.

3 Older milling machines for cemented carbide are often supplied with such low power that the rpm of the motor decreases during the top load. This can be relieved by flywheels.

Plastic Flow in a Lead Extrusion

By C. T. YANG[1] and E. G. THOMSEN[2]

Commercially pure lead was extruded through a sharp-edged circular orifice by an indirect or inverted extrusion process. Metal flow directions and plastic strains were determined during a small stepwise deformation process from distorted, originally square, grid-line network scribed on a meridian plane. It was found that the metal flow directions do not coincide with instantaneous grid-line directions nor with the streamline directions in frictionless or potential fluid flow. It was further found that the metal located in the corner, formed by cylinder wall and die face, was in a state of plastic deformation. This observation is contrary to previously reported investigations and shows that the surface quality of the finished extrusion is dependent on the surface quality of the billet from which the extrusion was pressed. A strain analysis was made from which principal stress directions in the plastic metal were calculated.

Nomenclature

The following nomenclature is used in the paper:

$\epsilon_x, \epsilon_y, \epsilon_z$ = normal infinitesimal plastic strains in the x, y, and z-co-ordinate directions, respectively

$\gamma_{xy}, \gamma_{yz}, \gamma_{zx}$ = infinitesimal plastic shear strains

θ = angles between principal direction and x-co-ordinate axis

β, β' = angles between intersecting grid lines before and after small additional plastic deformation, respectively

l, m = direction cosines with respect to x and y-co-ordinate axes, respectively; subscripts are used to identify a particular grid line.

Introduction

The extrusion of nonferrous alloys ranks among one of the most important forming processes today. It has inherent advantages over other processes such as rolling and forging in that complicated shapes with re-entrant angles can be obtained readily with this process and that the dies are relatively simple, easily manufactured, and replaced in an extrusion press, if it is desired to alter the shape of the product.

The process consists of placing a roughly fitting cylindrical billet in the cylinder of an extrusion press and forcing the metal through a steel aperture having the desired shape of the product; it is somewhat analogous to forcing tooth paste out of the orifice of a tooth-paste tube. The billet and the extrusion cylinder are heated to the extrusion temperature; this is normally above the rapid recrystallization temperature of the alloy and results in little or no work-hardening of the finished product.

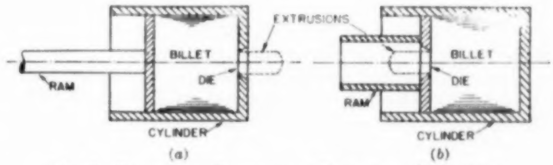

Fig. 1 Schematic Diagram of Extrusion Processes
(*a*, Direct extrusion: Ram and extruded part move in same direction. *b*, Indirect extrusion: Ram and extruded part move in opposite direction.)

Two methods of applying the extrusion pressure are commonly used and are illustrated schematically in Fig. 1:

1 Direct extrusion process, Fig. 1(*a*). The pressure is applied by means of the hydraulically operated ram at one end of the billet through a follower plate or piston. The forming pressure is transmitted through the billet in forcing the metal through the die at the other end of the billet.

2 Indirect or inverted extrusion process, Fig. 1(*b*). The pressure is applied through the die itself fixed to the hydraulically operated ram and the extruded product enters the hollow ram.

The direct extrusion process has the advantage that the length of the product is limited only by the size of the billet or the length of the runout table. Its disadvantage lies in the fact that the forming pressure, in addition to causing plastic flow in the metal, must be sufficiently high to overcome friction induced at the wall of the cylinder, since the billet moves relative to the cylinder. In the inverted or indirect extrusion process the cylinder-wall friction is reduced materially, since little relative motion exists between billet and cylinder wall, but the length of the extruded product is limited by the length of the hollow ram.

From both the scientific as well as technological point of view it is of interest to know the deformation mechanism and the resulting strain and stress distribution in the billet and finished product. A survey of the literature reveals that a number of investigators have analyzed the extrusion process analytically as well as experimentally. Van Iterson (1)[3] and Hill (2) have used Hencky's (3) plastic-sector method for finding the slip-line field and hence the plastic region of the material in the billet under conditions of plane strain. The material was assumed to be a plastic-rigid solid, which is rigid when stressed below the yield point, but will flow plastically without work-hardening, when the maximum shear stress within the body reaches a critical value. This value is $k/2$ in Tresca's (4) criterion while for von Mises' (5) it is equal to $\frac{k}{\sqrt{3}}$, where k is the yield strength in tension of the material under consideration. The error involved in idealizing the material is believed not to be serious, for actual metals and alloys above the rapid recrystallization temperatures employed in extrusion processes essentially have a flat plastic stress-strain curve and, in fully developed plastic flow, the effects of the small elastic strains are negligible.

Siebel (6), Sachs (7), and Unckel (8) have used the experimental approach to obtain the deformation pattern. Unckel has used a composite specimen in extruding an experimental billet into a solid rod and has concluded from his observations that the direction of flow agrees approximately with that of a frictionless fluid discharged through a sharp-edged orifice, i.e., potential flow.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, August 14, 1951. Paper No. 52—F-18.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Siebel and Sachs have scribed a square grid network on the meridian plane of sectioned round billets and have made their measurements on the deformed network after pressing. Siebel has calculated the principal strains and in turn from them the principal stress trajectories within the deformed billet. His strain analysis was based on an instantaneous deformation pattern of the grid network of the billet but he made the incorrect assumption that the material deforms along the grid lines originally parallel with the axis of the extrusion direction.

The present investigation has as its purpose a refinement on the experimental technique of obtaining deformation patterns within an extrusion and to calculate the principal stress trajectories from a more accurate strain analysis.

Experimental Technique and Results

The extrusion process selected for the present investigation was the inverted process in order to eliminate cylinder-wall friction as much as possible. Commercially pure lead was chosen for the billet material, since the recrystallization temperature of lead is at or near room temperature and thus permits atmospheric temperature pressing without inducing work-hardening. The cylinder of the extrusion press was $4^{5}/_{16}$ in. diam and the polished circular sharp-edged orifice had a diameter of $1^{3}/_{8}$ in. resulting in a ratio of reduction of approximately $^{1}/_{4}$. The meridian plane of the sectioned billet was provided with a machined square grid network spaced 0.10 in. apart and a line depth of 0.007 in. The billet was lubricated with a mixture of oil and white lead and was extruded at a slow speed of approximately 1 ipm for one third of its length. The billet and extruded rod were then removed from the press, cleaned, and photographed with oblique illumination in order to bring out the deformed grid lines.

Fig. 2 shows the deformation pattern of such an extrusion pressed from a machined casting in the as-cast condition. It is evident from this figure that it is unsuitable for precise measurements of the deformed grid network, for flow across the meridian plane obviously has taken place. Such crossflow possibly results from two reasons; namely, the scribed plane is not a meridian plane, or the metal is macroscopically inhomogeneous and anisotropic. The latter was found to be the case for the grains in the as-cast billet were relatively large and thus were causing heterogeneous flow. This condition could be cured readily by deforming the lead billet severely before machining and thus refining the grain size. The result of pressing the predeformed billet is shown in Fig. 3. Measurements of the deformed grid throughout the billet were made from the negative of this figure on a contour projector to a linear accuracy of 0.0001 in. and an angular accuracy of 1 min. After the measurements of the first pressing were completed, the billet was given a second pressing by reducing the length of the billet approximately $^{1}/_{4}$ in. The pattern of the deformed grid network of the second pressing had the appearance of that of the first. The deformed grid lines also were measured on the contour projector to permit an incremental strain analysis to be developed in the following section of this paper.

The results of the two pressings are shown graphically in the quarter section of the billet in Fig. 4. The circles of this figure give the position of intersecting grid lines after the first pressing and the end of the arrow heads the position of these same grid lines after the second pressing; the solid lines connecting these points indicate the approximate direction of flow. It may be noted that plastic flow occurs in the square corner formed between die and cylinder wall normally believed to contain a stagnant region, called dead metal.

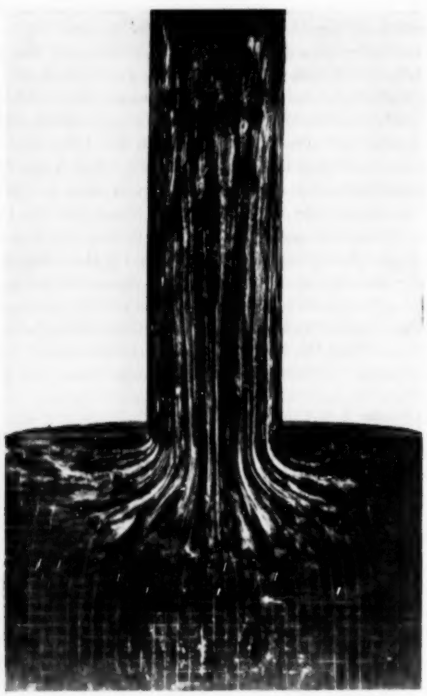

Fig. 2 Deformation Pattern During Indirect Extrusion of Square Grid Network on Meridian Plane of Lead Billet
(Structure of billet was in as-cast condition.)

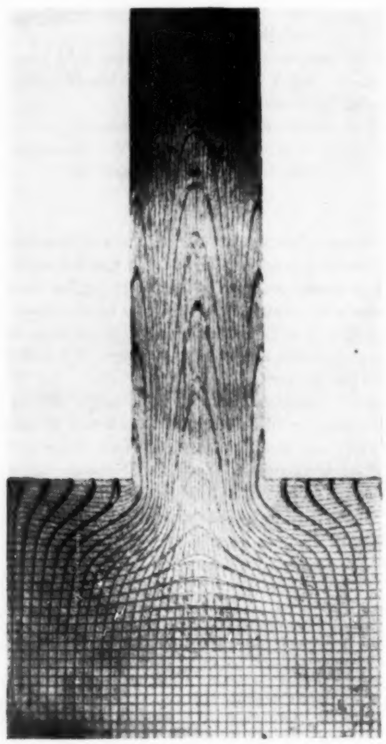

Fig. 3 Deformation Pattern, During Indirect Extrusion of Square Grid Network on Meridian Plane of Lead Billet
(Structure of billet was refined by cold-work and annealing.)

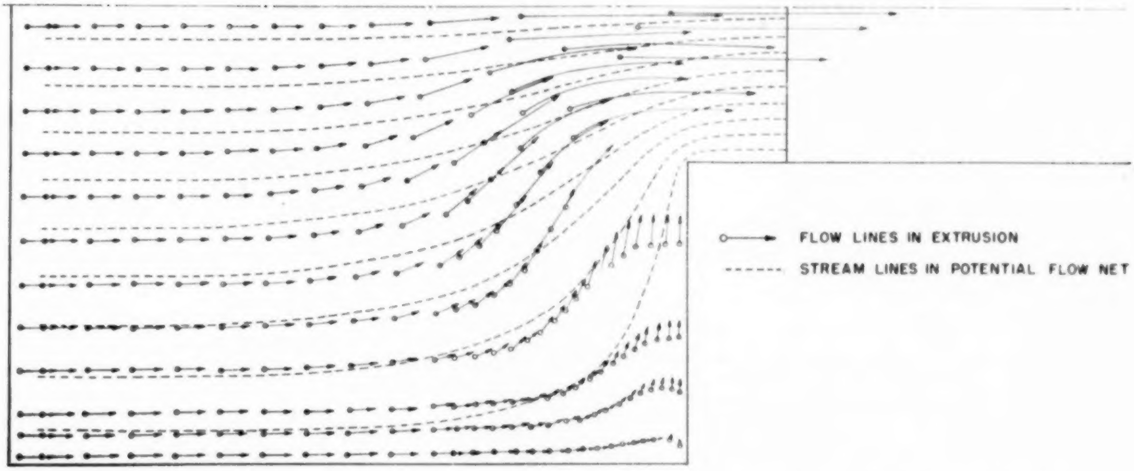

Fig. 4 Flow Directions on Meridian Plane of Lead Billet During a Stepwise Indirect Extrusion Process

In order to ascertain if a lubricating film permitted metal to flow in the dead-metal region, the extrusion press and an experimental billet were cleaned thoroughly of all traces of oil with carbon tetrachloride followed by pressing. It was found that metal flowed along the die surface and the deformation pattern was similar to that which was obtained when the billet had been lubricated. These observations are contrary to those reported in the literature.

The dotted lines superimposed on the metal flow directions in Fig. 4 are the stream lines in potential flow. These lines were obtained by an electrical-analog method; the electrolytic tank used had the same physical boundaries as the extruded billet. It is evident from this that the flow directions of the plastic metal do not agree with the stream lines for the fully developed plastic region.

Theory and Discussion

The state of strain at a point during an infinitesimal plastic deformation may be described completely by three normal infinitesimal strain components ϵ_x, ϵ_y, and ϵ_z in the x, y, and z-directions, respectively, and the three infinitesimal shear strains γ_{xy}, γ_{yz}, and γ_{zx}. For the case of deformation on the meridian plane of a cylindrical extrusion the shear strains γ_{yz} and γ_{zx} vanish because of axial symmetry. This can be verified readily by the fact that the meridian plane of the extrusion shown in Fig. 3 at any stage of the plastic deformation remains a plane. The three normal strains ϵ_x, ϵ_y, and ϵ_z are related through the condition of volume constancy by the equation

$$\epsilon_x + \epsilon_y + \epsilon_z = 0 \qquad [1]$$

from which ϵ_z can be determined in terms of the sum ϵ_x and ϵ_y. Thus the determination of the infinitesimal strains ϵ_x, ϵ_y, and γ_{xy} on the meridian plane will permit the complete determination of the infinitesimal strain components at any point during an infinitesimal plastic deformation. Furthermore, the principal infinitesimal strain direction may be obtained from ϵ_x, ϵ_y, and γ_{xy} from the well-known equation

$$\tan 2\theta = \frac{\gamma_{xy}}{\epsilon_x - \epsilon_y} \qquad [2]$$

where θ is the angle between the principal direction and the x-co-ordinate direction which may be chosen as the extrusion direction. If the material being extruded is isotropic and homogeneous, the infinitesimal principal strain and stress directions are coincident. The determination of the infinitesimal strain components ϵ_x, ϵ_y, and γ_{xy} at any point in the plastic mass are sufficient, therefore, to calculate the principal stress directions.

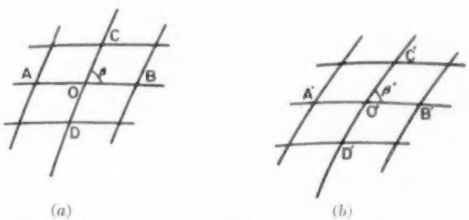

Fig. 5 Schematic Diagrams of Grid Lines in a Stepwise Extrusion Process

(a. Grid lines at any state of deformation. b. Grid lines after additional small deformation from state a.)

In order to calculate the principal stress directions, however, it is necessary to relate the infinitesimal strain components ϵ_x, ϵ_y, and γ_{xy} to the incremental grid-line deformation of the stepwise extrusion process. Consider the schematic intersecting grid lines of an extrusion as shown in Fig. 5. A point O given by two such lines of Fig. 5(a) changes into point O' of Fig. 5(b). The grid lines intersecting point O are bounded by other grid lines at small distances from O permitting assigning them a finite length A-B and C-D. During an incremental plastic-deformation process these grid lines transform to length A'-B' and C'-D', respectively, while any angle between intersecting grid lines such as β changes to β'. The normal finite plastic strains or natural strains along grid line A-B and C-D are given by $\log_e (A'B')/(AB)$ and $\log_e (C'D')/(CD)$, respectively, and the shear strain (9) by $\sin(\beta - \beta')$. If the deformation is small these strains may be equated to the total infinitesimal plastic strains, which may be given in terms of their components as follows (see Frocht (10), also Coker and Filon (11) for derivation of equations)

$$\left.\begin{aligned} \log_e \frac{A'B'}{AB} &= l_1^2\epsilon_x + m_1^2\epsilon_y + l_1m_1\gamma_{xy} \\ \log_e \frac{C'D'}{CD} &= l_2^2\,\epsilon_x + m_2^2\,\epsilon_y + l_2m_2\gamma_{xy} \\ \sin(\beta - \beta') &= 2\,l_1l_2\epsilon_x + 2\,m_1m_2\epsilon_y + \\ &\qquad (l_2m_1 + l_1m_2)\gamma_{xy} \end{aligned}\right\} \qquad [3]$$

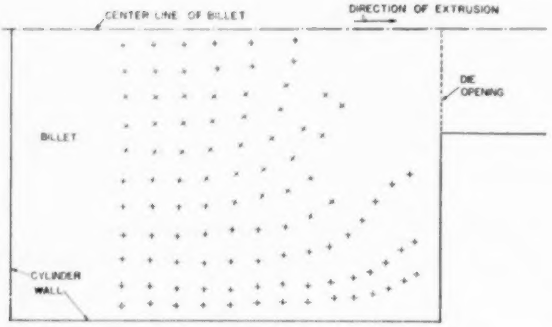

FIG. 6 DIRECTIONS OF PRINCIPAL STRESSES IN LEAD BILLET DURING INDIRECT EXTRUSION PROCESS

(Stress directions were calculated from strain analysis of deformed grid network on meridian plane of billet.)

The direction cosines l_1, m_1, and l_2, m_2 of Equations [3] refer to the orientation of length A-B and C-D with respect to the x and y-co-ordinate axes before the stepwise deformation was carried out. From measurements before and after deformation of any intersecting grid lines it is possible to evaluate the left side of Equations [3] and the coefficients of the strain components and, therefore, a simultaneous solution of Equation [3] yields the magnitude of components ϵ_x, ϵ_y, and γ_{xy}. The principal stress direction, under the previously mentioned assumptions of isotropy and homogeneity, may then be calculated by substituting the components ϵ_x, ϵ_y, and γ_{xy} into Equation [2].

The necessary calculations for the stepwise extrusion have been carried out and the results are shown as crosses in Fig. 6. The direction of one line of each cross indicates the maximum principal direction, while the other shows the minimum principal direction.

It was not possible to calculate the principal stress direction at all points of the plastically deformed billet. In the region farthest removed from the die the deformation of the grid lines was small and, consequently, did not permit accurate calculation of the principal stress directions. On the other hand, in the vicinity of the die opening, the plastic distortion of the grid lines during the small deformation process was so great that Equations [3] are no longer valid. Consequently, the determination of the principal stress directions was carried out only for a relatively narrow region. It is possible, however, to obtain accurate data for any of the other regions by suitably adjusting the extent of the deformation during a stepwise extrusion process.

It is of interest to note that the principal directions as obtainable by the slip-line or plastic-sector method mentioned previously are in approximate agreement with the present results. This comparison is given in Fig. 7. The solid lines are the faired stress trajectories, while the solid crosses are the principal stress directions from Fig. 6, and the dotted crosses the principal stress directions which were obtained from the slip lines in Fig. 8.

It is possible to obtain the stress distribution in the billet from the known strains, principal stress trajectories, and known boundary conditions. The solution requires integration of the equilibrium equations in cylindrical curvilinear co-ordinates along the principal stress trajectories. This integration has not been performed, because the principal stress trajectories are not known exactly but the method of solution has been obtained and will be presented in a future paper.

CONCLUSION

1 The extrusion of a solid cylindrical bar from a round billet yields a flow pattern at the meridian plane, which does not agree with that of potential flow.

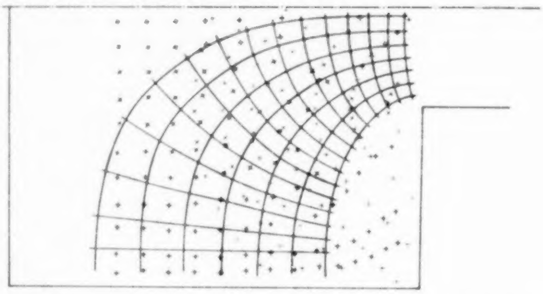

FIG. 7 APPROXIMATE PRINCIPAL STRESS TRAJECTORIES ON MERIDIAN PLANE OF LEAD BILLET DURING INDIRECT EXTRUSION PROCESS

(+ Principal direction from Fig. 6. + Principal directions from Fig. 8.)

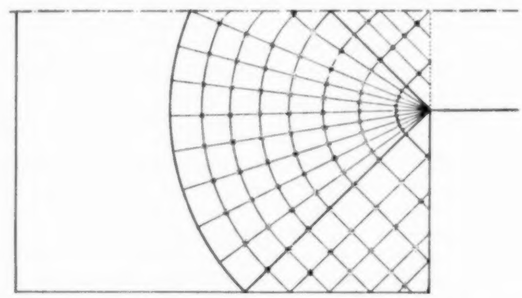

FIG. 8 TWO-DIMENSIONAL SLIP-LINE FIELD OF PLASTIC-RIGID SOLID DURING INDIRECT EXTRUSION PROCESS

(—Slip lines + Principal stress directions at 45 deg to slip lines)

2 The direction of flow cannot be determined from an instantaneous pattern of deformed grid lines, but requires an analysis of a stepwise extrusion process, for the direction of flow is not parallel to those grid lines originally aligned with the extrusion axis.

3 The stress trajectories calculated from the deformation of grid lines in a stepwise extrusion process agree approximately with those obtained from the plastic-sector or slip-line method.

4 For the conditions of the present investigation no stagnant or dead-metal region was observed when extruding lead with either lubricated or nonlubricated boundary surfaces.

5 The surface quality of the extruded shape depends on the quality of the surface of the billet from which it is pressed, because metal flow occurs along the solid die surface and not, as has been believed previously, along a natural die surface. In consequence of this flow mechanism, defects originally located in the surface of the billet will appear on the surface of the extrusion.

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Discussion

B. W. Shaffer.[4] This is an interesting study of the behavior of lead during the extrusion process. The figures, obtained by studying grid deformation, vividly show the flow pattern of particles during this process.

In Fig. 8 of the paper the authors analyzed the extrusion of a billet which is symmetric about the center line shown to the right of the figure. Therefore but one half of the process is shown. Nevertheless, because this center line is an axis of symmetry, all shear lines should intersect it at an angle of 45 deg. This condition, however, was not satisfied by the semicircular arc shown at the boundary between the rigid and plastic regions. Fig. 8, therefore, is incorrect and it cannot possibly be a solution to the problem being studied.

Three correct solutions to the extrusion process with $^1/_2$ reduction have been published. R. Hill[5] published two. Each of his solutions has a stagnant, or dead-metal region in the square corner between the die and cylinder wall, and each satisfies all boundary conditions. E. H. Lee[6] published a third solution to the same problem. His analysis, shown in Fig. 9 of this discussion, also satisfies all boundary conditions, but does not show a dead-metal region.

[4] Assistant Professor of Mechanical Engineering, New York University, New York, N. Y. Jun. ASME.

[5] "A Theoretical Analysis of the Stresses and Strains in Extrusion and Piercing," by R. Hill, *Journal of the Iron and Steel Institute*, vol. 158, 1948, pp. 177–185.

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The extrusion pressures required by these solutions were not the same. In fact, the one without a dead-metal region required the least pressure. Since flow will follow that pattern which requires the minimum applied force, extrusion with a $^1/_2$ reduction,

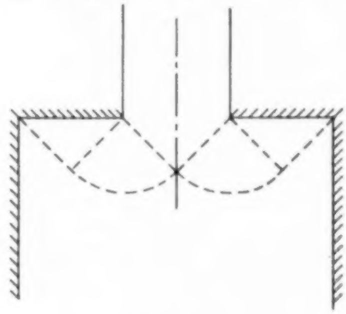

Fig. 9

according to these plane-strain solutions, will take place without a dead-metal region.

It is interesting to note that the authors did not observe a dead-metal region in their experiments.

Authors' Closure

The authors wish to thank Professor Shaffer for his discussion of the paper, and agree with him that Fig. 8 requires a minor modification. Lines should intersect the center line of the extrusion at an angle of 45 deg. This requires a slight shifting of a few lines and Fig. 8, except for this modification, is the correct solution of the inverted extrusion problem as given by Hill.

Additional experiments and calculations which have been made by the authors recently, indicate that the solution given in Fig. 9 does not agree with the experimental observations in spite of the fact that this solution permits plastic flow along the die surface. As a matter of fact, they find, that the plastic sector of Fig. 8 is a good first approximation for the maximum shear directions. They also observed that the shear strain rates in the so-called dead-metal region, i.e., outside of the plastic region of the Hill solution, are opposite in sign to those within this sector. These findings will be reported in a paper now in preparation.

Some Observations on Chip Curl in the Metal-Cutting Process Under Orthogonal Cutting Conditions

By R. S. Hahn,[1] Worcester, Mass.

An experimental study of the phenomenon of chip curl is made. It was found that transient behavior of the chip curvature was independent of cutting speed. Evidence is offered to show that chip curl originates in the shear region. Numerous hypotheses were made to explain the curl and were examined. It appears that the chip curl is associated with thermal effects in the shear region.

Nomenclature

The following nomenclature is used in the paper:

- q = heat-source strength
- v = temperature
- κ = thermal diffusivity
- K = thermal conductivity
- V = cutting speed
- V_x = component of V along shear plane
- V_z = component of V normal to shear plane
- t, T = time
- l = half length of heat source in direction of motion V_x
- b = width of heat source
- x, y, z = co-ordinates fixed in workpiece, x,y-plane parallel to shear plane
- $x'y'z'$ = moving co-ordinates parallel to x, y, z, $x,'z'$-plane coincident with shear plane
- $X = \dfrac{V_x x}{2\kappa}$
- L = Péclet number, $\dfrac{V_x l}{2\kappa}$
- τ = shearing stress
- J = mechanical equivalent of heat constant
- r = radius of curvature

Introduction

It is well known that chips, arising in the metal-cutting process with single-pointed tools, often exhibit a tendency to curl. To date there appears to be no satisfactory explanation of this phenomenon. Although the curling of chips as such is of little concern to those in the machine shop, except perhaps when it is desired to break the chip, this effect, when understood, may help in the theory of machining. The curvature of the chip is an additional quantity which may be measured and, when its meaning is understood, will provide a deeper insight into the metal-cutting process.

It often happens that when a study is made of one quantity, light is sometimes shed on other phenomena which may have scientific and commercial significance.

In the past, it appears that little work has been done on this aspect of metal cutting. Henriksen[2] has proposed that the chip is bent by the combined action of normal and tangential stress at the tool-chip interface. These stresses unquestionably influence the curvature but, as will be shown, are not of paramount importance in causing the chip to curl.

Experimental Setup

It will be observed if an interrupted cut is made under good cutting conditions, that a sequence of spiral chips is produced. In the case of orthogonal cutting on the end of an interrupted tube, each revolution of the work, under good cutting conditions, will produce a chip like that shown as A in Fig. 1. As the chip

A B C D

A, SAE 1020	600 fpm	831 Carboloy tool
B, "Marathon" Cr-Va steel	600 fpm	831 Carboloy tool
(2.22 C, 0.3 Mn, 0.34 S, 0.8 Mo, 12 Cr, 0.2 Va)		
C, Armco iron	600 fpm	831 Carboloy tool
D, Armco iron	600 fpm	883 Carboloy tool

Fig. 1 Chips From an Interrupted Cut Showing Transient Nature of Chip Curvature

is formed at the beginning of the cut it will have a small radius, i.e., large curvature. As the cut proceeds the curvature will reduce and approach more or less a steady-state curvature, usually taking several inches of tool travel to accomplish this. It is clear that some transient is occurring and reaching equilibrium.

In Fig. 2 a short tubelike workpiece (2 ft in circumference) is mounted on a boring head. A large angle bracket which carries

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[1] Consulting Engineer, The Heald Machine Company. Mem. ASME.

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 Manuscript received at ASME Headquarters, January 8, 1952. Paper No. 52—F-16.

[2] "The Stress Distribution in the Continuous Chip—A Solution of the Paradox of Chip Curl," by E. K. Henriksen, Trans. ASME, vol. 73, 1951, pp. 461–466.

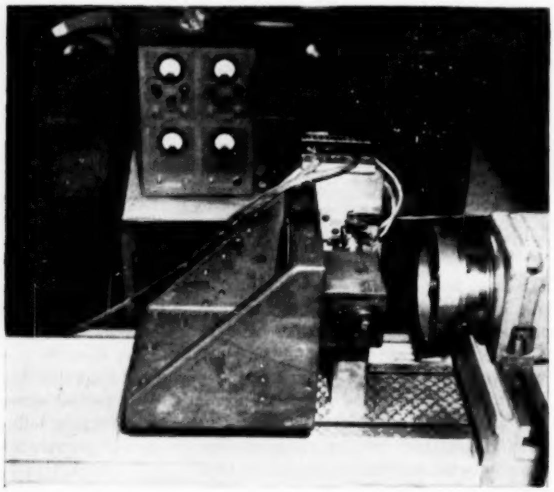

Fig. 2 Dynamometer for Recording Transient Cutting Forces

Fig. 3 Interrupted Workpiece With Insert for Interrupting Shear-Region Phenomena

a heavy block is mounted on the table of a precision boring machine. A cantilever bar with a tapered shank is drawn securely into the tapered hole in this block. A simple straight-edged tool is mounted near the free end of the bar in such a way as to produce orthogonal cutting on the end of the tube. Variable-reluctance magnetic gages operate on the far side of the tool and are just visible in Fig. 2. In back of the plate carrying the gages, a bushing with a large oil-filled clearance encircles the end of the bar, providing a terminal dashpot for vibration control. The gages, amplifier, and oscillograph, shown in the background, are capable of responding to about 1200 or 1500 cycles per sec (cps). The natural frequency of the bar is in the neighborhood of 700 cps with a spring constant at the tool point of 146,000 lb per in. The gages are used to measure the instantaneous cutting forces during a revolution of the workpiece.

Fig. 3 shows a typical workpiece. The surface to be cut is $^1/_8$ in. wide and extends upward in Fig. 3. A window or interruption is cut in the workpiece between the points marked "start" and "finish" to allow rapid positioning of the workpiece for engaging the tool so that one and only one uniformly deep cut is made in one revolution. By virtue of its holder, the workpiece is capable of a small endwise movement. It is "cocked" against heavy springs in its holder and when, during the rotation of the work, the interruption comes opposite the tool, the springs are triggered; the workpiece then moves toward the tool until it is restrained by a positive stop. It requires about 0.002 sec for the workpiece to be brought into cutting position. Once in cutting position the tool engages the work abruptly at the point marked start. Cutting then proceeds at a uniform depth of cut until the interruption again reaches the tool at the point marked finish. During the time the tool is in the interruption, another spring (not shown in Fig. 2) is triggered, which rapidly deflects the tool backward away from the work by an amount greater than the tool deflection during the cut, so that although the workpiece continues to rotate, no subsequent cutting is done. In this way a uniformly deep cut is taken at reasonably high speeds (600 fpm) and the cut surface of the workpiece is preserved for observation. The interruption is approximately 4 in. long leaving about 20 in. for the length of cut.

In addition, in order to get a comparison between chips which were formed with a tool that was cool and had an adsorbed film, and chips that were formed by a tool cutting in the steady state (no adsorbed film and up to temperature), a narrow radial slot $^1/_{16}$ in. wide is made in the workpiece about $4^3/_4$ in. from the starting point, and a strip of metal similar to the workpiece is driven tightly into this slot. After a preparatory machining this insert appeared practically invisible, nevertheless, later on, it will be seen to provide an effective interruption for "shear-region phenomena" without disturbing appreciably "tool-face phenomena." The position of the insert is indicated in Fig. 3.

In the later tests the setup also was provided with means for recording the transient temperature changes at the tool-chip interface. The workpiece and mounting were insulated and a connection to the tool-work thermocouple was made through a disk running in a mercury trough. The other connection was simply a wire fastened to the tool shank. This caused no trouble because no bulk heating of the tool tip or shank occurred during the interval of cutting (about 0.02 sec). No attempt was made to calibrate the tool-work thermocouple because the entire interest was in transient behavior at the tool-chip interface.

The tool-work thermocouple was connected to a direct-coupled amplifier and thence to one channel of the two-channel oscillograph. Because of the availability of only two channels in the oscillograph, tests had to be made with temperature and thrust force on the oscillograph and then repeated with thrust and cutting force on the oscillograph. However, this procedure is quite satisfactory as conditions are reasonably reproducible.

Experimental Results

Fig. 1 shows some typical chips. There are two chips for each test, a short one, corresponding to the cutting interval between the start of the cut and the insert, and a long one corresponding to the interval between the insert and the end of the cut. The small end of the spiral in each case is formed first and it is clear that the radius of curvature increases as cutting proceeds (the curvature decreases), gradually approaching approximately a steady state. In discussing these chips, the curvature rather than the radius of curvature has fundamental significance as will become apparent. Changes in the radius of curvature when the radius is large produce trivial changes in curvature; hence, although there is some fluctuation in radius after the "steady state" has been reached, in terms of curvature it is very small.

Fig. 4 shows the curvature plotted against distance traveled for two different materials. It is seen that the distance traveled to reach equilibrium is different for different materials and also the steady-state curvature is different. Consequently we have two characteristics to focus attention on, i.e., the distance traveled to reach equilibrium, or some fraction thereof, and the steady-state curvature.

Fig. 5 illustrates a typical oscillogram of the thrust and cutting forces taken during one revolution of the workpiece. The upper trace at the left-hand end is seen to deflect abruptly downward when cutting starts. This trace measures the thrust or tool-face frictional force. The lower trace measures the cutting force. The "pip" about 1/4 the distance along the active section corresponds to the insert. Note the well-damped wave and fast response where the tool drops into the interruption at the end of the cut.

Fig. 6 illustrates the effect of cutting speed on chip curvature. It may be seen that as the speed is changed nearly 30:1 the distance traveled in all cases is essentially the same.

The effects of rake angle between −20 deg and +20 deg on chip curl were not pronounced. The effect of depth of cut, varying between 0.001 in. and 0.005 in., was to change the curvature, the larger depths producing less curved chips, without affecting appreciably the distance traveled to reach equilibrium. Since these effects do not contribute to the argument to follow, details are omitted for brevity.

Various tool materials were tested, and it was observed that those producing higher friction tended to yield straighter chips. This is illustrated in Fig. 1 by the chips labeled C and D. The high-friction tool produced the D chip.

Fig. 7 shows an oscillographic record of the tool-chip interface temperature. The upper trace at the left-hand end as before is the thrust force. The lower at the left-hand end represents the interface temperature. Note the very slight radius of the temperature trace as it approaches equilibrium and the sharp "drop" when the tool drops into the interruption. The distance traveled for the temperature to reach equilibrium is about 0.130 in. along the workpiece.

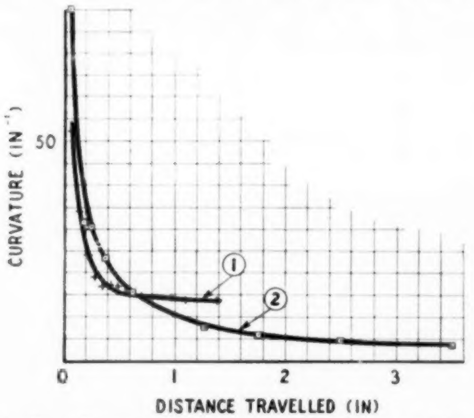

Fig. 4 Chip Curvature Plotted Against Distance Traveled Along Workpiece
(Curve 1, "Zimca" bronze. Curve 2, SAE 1020.)

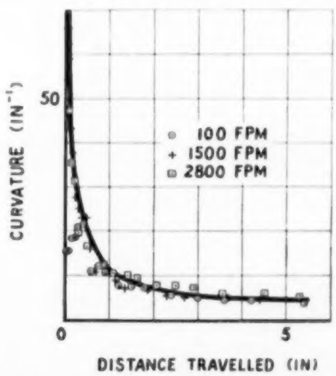

Fig. 6 Chip Curvature Versus Distance Traveled Along Workpiece for SAE 1020 at 100, 1500, and 2800 Fpm

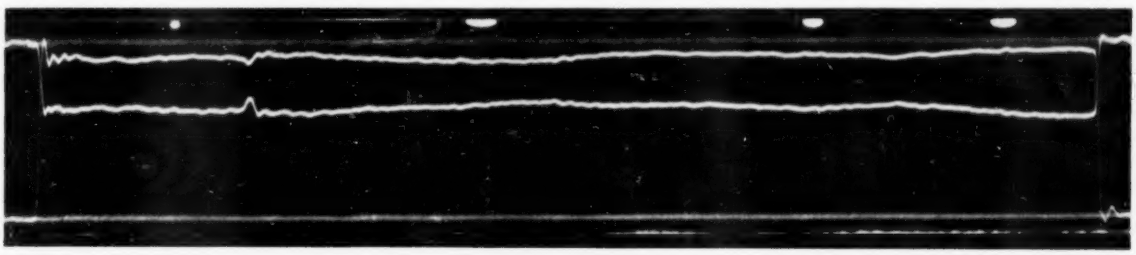

Fig. 5 Oscillogram Showing Thrust and Cutting Forces
(SAE 1020; 0.003 depth of cut, 600 fpm, 831 Carboloy, 0 deg rake, 12 deg clearance.)

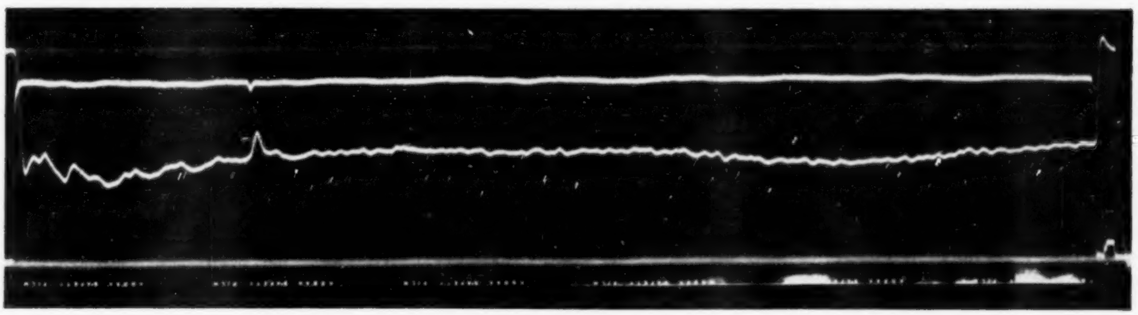

Fig. 7 Oscillogram Showing Tool-Chip Interface Temperature
(SAE 1020; 0.003 depth of cut, 600 fpm, Kennametal K5H, 0 deg rake, 12 deg clearance.)

Theory of Chip Curl

Tool-Chip Interface Effects. In view of the test results cited let us consider what might cause such behavior.

Since the spiral chips can be dissolved in acid or heated to incandescence without change of curvature, it would seem that residual stresses could be eliminated as a cause of chip curl.

If we suppose the chip to be bent by the action of the forces at the tool interface, it would seem as though the bending moment about the shear region would be greatest for straight chips which presumably bear a greater distance up the tool face than tightly coiled chips which contact the tool for a lesser distance. Consequently, where the curvature is greatest the moment is least and where the curvature is least the moment is greatest, which is just the reverse of what is needed. Furthermore, in view of the constancy of the forces in Fig. 5, it would be difficult to explain the transient behavior in terms of bending moment. It is true however as mentioned earlier, that high tool-face friction tends to cause less curvature but the tendency to curl is inherent and is only modified by friction. This hypothesis then may be eliminated.

It might be supposed that the change in curvature shown in Fig. 4 or Fig. 6 could be correlated with the build-up of tool-chip interface temperature. Since it takes several inches of travel to reach chip-curl equilibrium (this distance corresponds to about $^1/_{10}$ the length of the "active" section in Fig. 7), and only about 0.130 in. of travel for the tool-chip interface temperature to reach equilibrium, it would appear that the interface temperature is not the controlling factor.

As another hypothesis as to the cause of chip curl, it might be supposed that the curvature is due to the built-up edge and that changes occur in the built-up edge during the first inch or two of travel. It might be mentioned at this point that during the course of 30 or 40 experiments most of the cutting was done under conditions supposedly free of a built-up edge; i.e., the chips were often clean and smooth-backed. The back of the chips often had a mirror finish. In some cases they had a matte finish. Regardless of whether the finish was mirrorlike or matte the spiral characteristic often was maintained.

Intermediate cases were observed where various fractions of the area of the back of the chip where highly burnished. In some instances, as a mirrorlike chip left the workpiece and tool at the end of the cut, it carried with it a very small built-up edge. These observations led to the conclusion that a small built-up edge existed in all cases. Furthermore, it was observed that the beginning of the first chip (where the tool is cold and lubricated) was very tightly coiled and always had a mirrorlike finish for about 0.050 in. along the chip (approximately 0.100 in. along the workpiece). Further along the chip occasional imperfections were observed. Consequently, it appears that the built-up edge is formed in the first 0.100 in. of travel. The second chip, formed at the insert, did not show this effect.

In view of the foregoing it seems reasonably clear that the built-up edge is formed in approximately the same time that it takes for the tool-chip interface temperature to build up, i.e., in about 0.100 in. of travel. Therefore chip-curvature effects do not appear to be related to the built-up edge.

Up to this point various tool-chip interface effects have been considered. In view of the fact that essentially the same spiraling effect is observed at the insert and the beginning of the cut and that the forces and interface temperature are not changed materially at the insert, it would seem that the chip curl is a phenomenon associated with the shear region and not the tool-chip interface region.

Shear-Region Effects. The behavior shown in Fig. 6, namely, that a constant distance traveled is required to attain equilibrium rather than a constant time, as the cutting speed is varied, suggests that the phenomenon is thermal in nature. Also, it often will be observed that the colored oxide film on the chip correlates with the curvature.

It is known from the theory of heat conduction that moving sources of heat, whose source strengths, q, are proportional to the velocity, require approximately a constant distance traveled to reach equilibrium, as the speed is changed. Fig. 8, derived for a

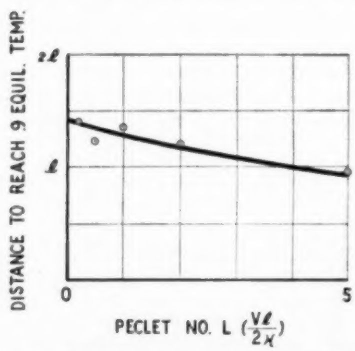

Fig. 8 Distance Traveled of a Square Heat Source Moving on Surface of a Semi-Infinite Solid, to Reach 0.9 Equilibrium Temperature Versus Peclet Number

(l = half-length of source; Peclet number $L = vl/2\kappa$.)

square heat source moving over the surface of a solid, from the work of Jaeger,[3] shows the distance traveled to reach 0.9 equilibrium temperature plotted against the Peclet number L.[4] The theory of moving heat sources gives rise to a dimensionless temperature of the form

$$\frac{\pi K V_s}{\kappa q} v$$

where the symbols are defined in the nomenclature. It has been shown[5] that V_s/q, a factor in the dimensionless temperature, is proportional to the shearing stress τ.[6] Therefore, assuming for the moment a constant stress τ, the effect of increasing velocities is felt only in the Peclet number L. Based on this, Fig. 8 was derived from Jaeger's[3] Fig. 3. From Fig. 8 it will be seen that the distance traveled to reach equilibrium is nearly constant and of the order l, the half length of the heat source. It may be of interest here, to note that the distance traveled by the chip over the tool face is about 0.050 in. to reach equilibrium temperature, as indicated in Fig. 7, and this, although somewhat greater than the half length of the chip-tool-interface source, is, nevertheless, of the order to be expected.

Before considering in detail thermal effects in the shear region, it appears that, since curvature effects are due to shear-plane phenomenon, the chip curvature is a measure of the velocity gra-

[3] "Moving Sources of Heat and the Temperature at Sliding Contacts," by J. C. Jaeger, Journal and Proceedings of the Royal Society, NSW 76, 1943, pp. 203–224.

[4] "A Remark About Standardization and 'A Dimensionless Analysis of Metal Cutting,'" by Andre Martinot—Lagrade, *Journal of Applied Physics*, vol. 21, 1950, p. 1197.

[5] "On the Temperature Developed at the Shear Plane in the Metal Cutting Process," by R. S. Hahn, Proceedings of the First National Congress of Applied Mechanics (to be published).

[6] The source strength q is obtained by dividing the shearing power (shear force, F_s × velocity of shearing V_s) in heat units, by the area of the shear plane A_s. Thus

$$q = \frac{F_s V_s}{A_s J} \quad \text{hence} \quad \frac{q}{V_s} = \frac{\tau}{J}$$

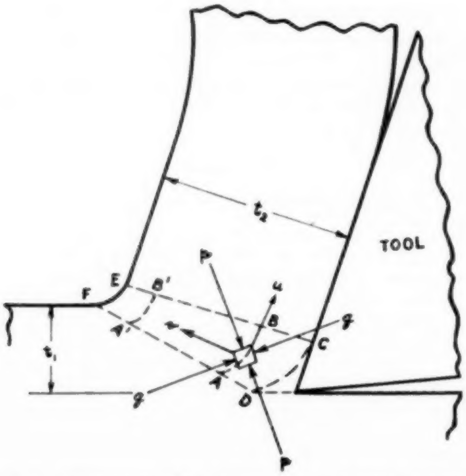

FIG. 9 ORTHOGONAL SLIP LINES u AND v IN SHEAR REGION

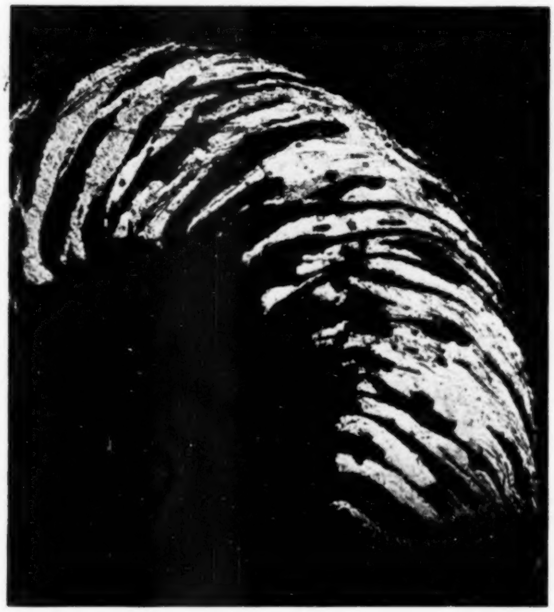

FIG. 10 PHOTOMICROGRAPH OF CHIP NEAR ITS BEGINNING SHOWING CURVED MAXIMUM CRYSTAL ELONGATION LINES; ×300

dient at the upper elastic-plastic boundary E-C in Fig. 9. Since the velocity at C is greater than at E, it follows that the chip will be curved and, assuming the velocity of chip flow u to change uniformly across the chip, the curvature is related to the change in velocity by

$$\frac{1}{r} = \frac{u_c - u_e}{u_e t_2} = \frac{\Delta u}{t_2 u}$$

This, of course, is true only when deformation due to tool-face friction is negligible. This deformation, called secondary flow by Cook and Shaw,[7] appears to be unimportant in the tests run here as may be seen by comparing the photomicrographs in Figs. 10 and 11. Remembering that the friction force and tool-chip temperature are constant, Figs. 5 and 7, it may be concluded from Fig. 11 which shows the chip after the steady state has been reached, and Fig. 10 which was taken near the beginning of the cut, that the secondary flow in this case is negligible, and therefore the curvature is a measure of the velocity gradient across the thickness of the chip.

The dimensionless steady-state shear-plane temperature distribution has been found[6] by considering the temperature developed by a uniformly distributed band heat source moving obliquely through an infinite solid. The dimensionless steady-state shear-plane temperatures are plotted over the shear plane in Fig. 12 for a shear angle of 20 deg and for various values of L. The abscissas $-l$ and $+l$ correspond to the ends of the shear plane terminating at the tool face and free surface, respectively. It will be seen that the peak in shear-plane temperature occurs not at the center of the shear plane, but toward the tool face. Therefore it may be expected that material in the region A-B in Fig. 9 will be at a higher temperature than that in the region $A'B'$. If we draw the principal stresses at 45 deg to the conventional shear plane, it appears that flow may take place, simultaneously along the orthogonal u, v slip lines. Thus it is the differential flow in the u-direction along E-C that causes curvature.

FIG. 11 PHOTOMICROGRAPH OF CHIP AFTER EQUILIBRIUM HAS BEEN ESTABLISHED

(Note straightness of crystal elongation lines and freedom from secondary flow; ×300.)

Fig. 12 giving the steady-state temperature distribution might serve as an explanation of chip curl. If so, then the transient temperature distributions calculated at times earlier than the steady state must be more unsymmetrical in order to correlate with the large curvature shown by the chips in the early stages.

The transient shear-plane temperature distributions are derived in the Appendix and several sample distributions are shown plotted over the shear plane in Fig. 13. At early times the temperature distribution is seen to be more symmetrical and not less. Therefore it may be concluded that this hypothesis, based on the assumptions made in the theory, is not the essential cause of chip curl. Furthermore, the steady state is practically reached when

$$\frac{V_s^2 T}{2\kappa} = 30$$

which corresponds to about 0.010 in. travel at 600 fpm and not several inches as observed from the chips.

It would be well to consider at this point some of the assump-

[7] "A Visual Study of Cutting," by N. H. Cook and M. C. Shaw, ASME Paper No. 51—A-73.

tions made in the theory of moving heat sources. Two important assumptions are as follows:

1 The heat source is taken to be uniformly distributed over a plane.

2 The heat-source strength q is taken to be independent of the temperature developed.

The deformation undoubtedly does not occur strictly in a plane, and the heat-source strength q is not necessarily uniformly distributed. Furthermore, since the heat-source strength is proportional to the product of velocity and shearing stress,[6] and since the shearing stress is dependent on temperature, it is clear that the source strength varies also with the temperature developed.

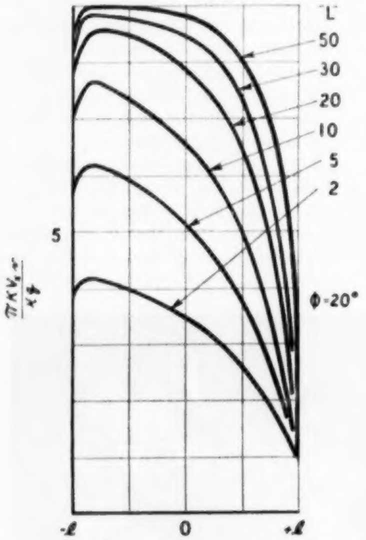

Fig. 12 Dimensionless Steady-State Shear-Plane Temperature Distribution

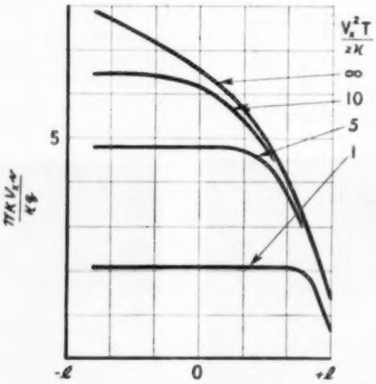

Fig. 13 Dimensionless Transient Shear-Plane Temperature Distribution

Although the problem of considering a heat source, nonuniformly distributed over C-D-E-F in Fig. 9 and whose strength is a function of temperature and stress, will not be attempted here, a few qualitative remarks may be made.

If we replace the uniform plane source in the foregoing theory by a source nonuniformly distributed throughout the plastic region C-D-E-F in Fig. 9 but still independent of temperature, it would seem that the fundamental result of a more symmetrical temperature distribution, followed by a more unsymmetrical distribution as the steady state is approached, would still occur. Therefore, in order to explain the phenomenon of chip curl, it is necessary to consider source strength or stresses that depend on the temperature. Thus we are led to a difficult problem.

Qualitatively, there are two possible mechanisms which can explain the transient characteristic of the chip curvature. (1) The source strength initially would presumably be uniform in the v-direction and, because of the higher temperatures developed near the tool face, would then become nonuniform, resulting in a reduction of strength in the high-temperature region. The tendency would be to produce in time a more uniform temperature and consequently less curvature. (2) The source initially could be distributed over the plastic region C-D-E-F, but after a short time higher temperatures would exist along E-C and cause a redistribution of heat source strength along the u-direction, thereby causing the plastic region to become progressively thinner. It should be clear that either or both of the mechanisms mentioned would lead to a large initial chip curvature, tending toward a smaller steady-state curvature. It also seems, intuitively, that the distance traveled to reach equilibrium would be much greater and therefore in better agreement with the experimental observations.

Conclusion

From the work presented here the following conclusions may be drawn:

1 It appears that the chip-curl phenomenon is associated with the shear region and not the tool-chip interface region.

2 The chip curvature is a measure of the u velocity gradient in the direction E-C in Fig. 9 when secondary flow is negligible.

3 In order to explain the transient effects regarding chip curl, it appears necessary to consider the problem connecting the flow stress of the material with the temperature developed.

Although the arguments presented here have been of a negative nature, it is hoped that some small amount of helpful information may be derived by the reader. It is believed that the transient thermal effects discussed are intimately connected with metal-cutting chatter, a type of self-excited vibration which often occurs in the machine shop.

Acknowledgment

The author wishes to acknowledge the generous support of the management of The Heald Machine Company for making it possible to carry out these experiments.

Appendix

Transient Temperature Distribution on Shear Plane

From the theory of heat conduction it is shown that the temperature at a point (x, y, z) at time t, in an infinite solid initially at zero temperature, due to a quantity of heat Q instantaneously liberated at the point $(x'y'z')$ at zero time is[3]

$$\frac{Q\kappa}{8K(\pi\kappa t)^{3/2}} \exp\left[-\frac{(x-x')^2+(y-y')^2+(z-z')^2}{4\kappa t}\right]$$

The temperature due to a rectangular array of point sources arranged in a plane parallel to the x, y-plane and moving obliquely relative to the x and z-axes may be found by integrating

$$dv = \frac{\kappa q dx'\, dy'}{8K[\pi\kappa(T-t)]^{3/2}} \exp$$
$$\left[-\frac{\{x-[x'-V_x(T-t)]\}^2+(y-y')^2+[z+V_z(T-t)]^2}{4\kappa\,(T-t)}\right]$$

over the area of the source.

If it be assumed that the temperature distribution on and ahead of the shear plane in the metal-cutting process is the same as that due to a rectangular heat source moving obliquely through an

infinite solid, then the shear-plane temperature distribution at the time T is

$$v = \frac{\kappa q}{8K(\pi\kappa)^{3/2}} \int_0^T \frac{dt}{(T-t)^{3/2}} \int_{-l}^{+l} dx' \int_{-b}^{+b} dy' \exp$$
$$\left[-\frac{\{x - [x' - V_x(T-t)]\}^2 + (y - y')^2 + [z + V_z(T-t)]^2}{4\kappa(T-t)} \right]$$

Carrying out the integration and letting $b \rightarrow \infty$ to get a band source and setting $z = 0$ to get the temperature on the shear plane gives

$$\frac{\pi K V_x}{\kappa q} v = \frac{1}{2}\left(\frac{\pi}{2}\right)^{1/2} \int_0^{\frac{V_x^2 T}{2\kappa}} \frac{du}{u^{1/2}} e^{-\left(\frac{V_z}{V_x}\right)^2 \frac{u}{2}}$$
$$\left\{ \operatorname{erf} \frac{X + L + u}{(2u)^{1/2}} - \operatorname{erf} \frac{X - L + u}{(2u)^{1/2}} \right\}$$

Evaluating the integral graphically for a shear angle of 16.7 deg ($V_z/V_x = 0.3$) and $L = 10$ and plotting against $(V_x^2T)/(2\kappa)$ for various positions along the shear plane, permits the transient dimensionless shear-plane temperature distribution to be plotted over the shear plane as shown in Fig. 13.

Discussion

B. T. Chao[8] and K. J. Trigger.[9] The author is to be commended for this thought-provoking paper on his observations of chip curl during machining operations. The ingenious method of isolating shear-zone behavior from that at the tool-chip region is an interesting experimental arrangement.

The vast differences in the curvature of chips C and D, Fig. 1 of the paper, suggest that tool-chip friction is an important factor in the over-all consideration of chip-curl phenomena. However, as shown in Figs. 5 and 7, the tool forces and, consequently, the tool-chip friction evidently attain substantially constant values almost immediately after cutting is started even though the chip curvature continues to decrease for some time. The contention that chip curl for a "given toolwork pair" is associated primarily with thermal effects in the shear zone and not tool-chip friction per se is in agreement with the writers' thoughts on the matter.[10]

Referring to Fig. 4 of the paper, it is noted that the chip for the bronze alloy attains a steady-state curvature in considerably less distance (about $^1/_3$) than does the 1020 steel. While the writers do not have the composition of the particular bronze used, many such alloys have a thermal diffusivity of the order of $2^1/_2$ to 3 times that of 1020 steel. Would not this variation in diffusivity contribute significantly to the differences in time necessary to reach steady-state curvatures?

In this paper and in an earlier paper[11] presented by the author, the temperature distribution along the shear plane was calculated by applying the theory of moving sources of heat in an "infinite" medium. With orthogonal cutting in which the chip formation takes place at the end of a tubular workpiece, it is clear that the foregoing condition does not exist. A better approach is to consider the workpiece to be only semi-infinite with an oblique band source moving along the surface as illustrated in Fig. 14 of this discussion.

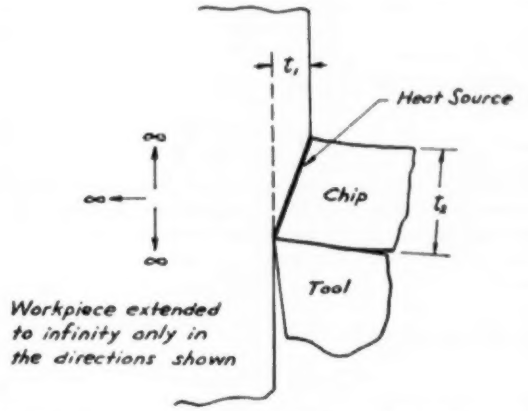

Fig. 14 Heating of a Semi-Infinite Workpiece in Orthogonal Cutting

Another important point neglected by the author is the fact that only a fraction of the energy liberated at the shear zone due to main chip shear goes into the workpiece by conduction. A major portion convects into the chip as a consequence of its motion. As defined by the author, the shear-plane-source strength q is obtained by dividing the shear power (shear force, $F_s \times$ velocity of shearing, V_s) in heat units by the area of the shear plane, A_s, i.e., $q = (F_sV_s)/(A_sJ)$. Thus the quantity q so defined represents the "total quantity of heat liberated" at the shear zone per unit time per unit area of the shear plane. Such shear-zone heat will go entirely into the workpiece only if the chip material were a perfect thermal insulator. Obviously, this is not the case and the use of the same q in Figs. 12 and 13 of the paper therefore is not correct.

The problem of temperature distribution along the shear plane has also been studied by the writers recently and results have been published elsewhere.[12] When turning spheroidized SAE 52100 steel at a feed of 0.0074 ipr with a triple-carbide tool of negative 2-deg rake, the temperature distribution at cutting speeds of 811 fpm and 116 fpm varies according to curves shown in Fig. 15 of this discussion. These curves were calculated on the basis of a "semi-infinite" workpiece. The quantity β represents the fraction of the shear-zone heat which conducts into the workpiece. It can be determined approximately by a method similar to that first used by Blok[13] in his solution of the temperature distribution in a friction slider. If β is the fraction of total shear-zone heat which passes into the workpiece, the fraction which leaves with the chip will be $(1 - \beta)$. Thus the average temperature rise of the chip as it leaves the shear zone is

$$(\Delta_s)_c = \frac{(1-\beta)q2l}{tV_sc\rho} \quad \ldots\ldots\ldots\ldots [1]$$

where

$2l$ = width of shear plane
t = feed

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[9] Professor of Mechanical Engineering, University of Illinois. Mem. ASME.

[10] "The Significance of the Thermal Number in Metal Machining," by B. T. Chao and K. J. Trigger, Trans. ASME, vol. 75, 1953, pp. 109–120.

[11] Reference (5) of the paper.

[12] Reference (3) of the paper.

[13] "Theoretical Study of Temperature Rise at Surfaces of Actual Contact Under Oiliness Lubricating Conditions," by H. Blok, Proceedings of the General Discussion on Lubrication and Lubricants, The Institution of Mechanical Engineers, London, England, vol. 2, 1937.

Other quantities are defined under Fig. 15 of this discussion. Since

$$q = \frac{\tau \epsilon V_c t}{2Jl} \qquad [2]$$

where ϵ is the unit shear strain, we have

$$(\Delta v)_s = \frac{(1-\beta)\tau\epsilon}{cJ\rho} \qquad [3]$$

The average temperature across the shear plane can be found by measuring the area under the curve using a planimeter. For a cutting speed of 811 fpm, the dimensionless temperature

$$\left(\frac{\pi}{2}\,\frac{c\rho J}{\tau}\,\frac{\Delta v}{\beta}\right)$$

has been found to be 8.4. Or

$$(\Delta v)_{\text{mean}} = \frac{16.8}{\pi}\,\beta\,\frac{\tau}{cJ\rho} \qquad [4]$$

By equating Equations [3] and [4], we finally have

$$\beta = \frac{1}{1 + \dfrac{5.35}{\epsilon}} \qquad [5]$$

Under the specified cutting conditions, the shear angle ϕ was found to be 22.8 deg and ϵ = 2.84. It follows therefore that β = 0.347.

For SAE 52100 steel, τ = 100,000 psi, c = 0.128 Btu/lb-deg F, ρ = 0.281 pci; $(\Delta v)_s$ as given by Equation [3] or [4] is 552 F. With an ambient temperature of 79 F, the average temperature of the chip as it leaves the shear zone is, therefore, 631 F. Considering all the necessary approximations introduced in the calculation, the agreement with the actually measured temperature value is good.

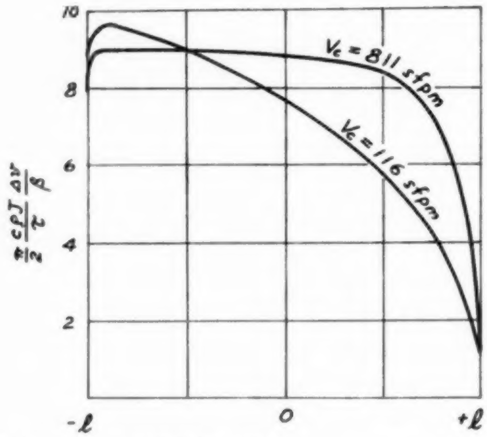
Even from the standpoint of "relative" temperature distribution, Fig. 12 of the paper is misleading since in the machining of steel the shear angle generally increases as cutting speed is increased. As a consequence, one gets the erroneous impression that the average temperature rise along the shear plane increases rapidly with speed—which is contradictory to observed facts. It is the writers' opinion that Fig. 15 of this discussion gives a better comparison since both the change of shear angle and the change of the length of the shear plane are considered there.

In reviewing the assumptions made in the theory of a moving heat source, the author lists two requirements which might not be fulfilled in the actual cutting process. A third factor, which is equally important, is the change of the temperature-dependent properties, such as specific heat and thermal conductivity, of the material as it is heated by chip shear.

Dimitri Kececioglu.[14] The writer deeply appreciates the extensive efforts of the author to find the causes of chip-curl transiency. All the factors investigated are well taken and their effects are as concluded in the paper. The writer would like, herewith, to attempt to throw a little more light on this complex problem. Among others, the following three facts appear to be of importance in this problem:

1 Owing to the sudden impact involved, as soon as the tool bites into the workpiece, the cutting speed would decrease momentarily and then pick up to a constant cutting speed as the cutting proceeds.

[14] Research Engineer, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Jun. ASME.



Work material = SAE 52100, 187 Bhn
Tool material = triple carbide
Depth of cut = 0.102 in.
Tool rake = negative 2 deg
Feed = 0.0075 ipr

In the figure:
V_c = cutting speed, sfpm
c = specific heat, Btu/lb-deg F
ρ = weight density, pcf
J = mechanical equivalent of heat
τ = dynamic shear stress, psi
Δv = temperature rise at shear plane above initial uniform temperature of workpiece
β = fraction of shear-zone heat which conducts into workpiece

Fig. 15 Dimensionless Steady-State Shear-Plane Temperature Distribution

2 The tool and the workpiece temperature would change from ambient to higher steady-state temperatures.

3 The tool-chip interface would undergo metallurgical and chemical changes and reach a steady state after a short period of cutting.[15,16]

These facts may have the following effects on the tool-chip interface and the shear zone:

1 At the beginning of the cut the coefficient of friction would be high owing to the reduced speed, but as the cutting speed approaches a steady-state value the coefficient of friction would assume a much lower steady-state value. It has been found that the coefficient of friction can undergo large changes in acceleration periods of even less than $^1/_{1000}$ sec.[17]

As a result of this change in the coefficient of friction, the thrust and cutting forces would have higher values at the beginning of the cut but assume a lower value at steady state. The higher forces create a higher stress concentration (shear and normal stress) on the shear zone nearer the tool edge than at steady state. This, in turn would cause a deeper stress penetration into the workpiece nearer the tool edge than close to the workpiece surface.

This greater stress gradient along the shear zone at the beginning of the cut would tend to give a wedge-shaped shear zone with a larger wedge angle than at steady state. The successive piling

[15] "Some Factors Affecting Wear on Cemented Carbide Tools," by E. M. Trent, *Machinery*, London, England, vol. 79, November 8, 1951, pp. 823–828; vol. 79, November 15, 1951, pp. 856–869.

[16] "Radioactive Cutting Tools for Rapid Tool-Life Testing," by M. E. Merchant, H. Ernst, and E. J. Krabacher. Published in this issue, pp. 549–559.

[17] "Chip Formation, Friction and Finish," by H. Ernst and M. E. Merchant, The Cincinnati Milling Machine Company, Cincinnati, Ohio, 1940, 48 pp.

up of such larger-angle wedges at the beginning of the cut, therefore, would cause the chip to come out more tightly curled than at steady state. This process of chip curling bears out the often quoted statement that the chip is "born curled."

The fact that no appreciable force changes were observed by the author during the tests (Fig. 5 of the paper) is misleading because the tool is immersed in a dashpot. Although this does damp out tool vibration satisfactorily, it also obscures the short-time force variations at the beginning of the cut.

The deeper stress penetration at lower cutting speeds is substantiated by the fact that the surface deformation on the workpiece is deeper at lower than at higher speeds and that the change of surface deformation with speed is more pronounced at the lower speed range.[18]

2 Indications are that shear-zone curvature undergoes changes from the time a cut is started until steady state is reached.[17] At the beginning of the cut the shear zone is concave upward but when steady state is reached, it becomes concave downward. When it is concave upward, the tool-chip contact length is longer. This additional length of contact (besides increasing the cutting and thrust forces) would tend to increase the bending moment on the chip and cause a greater curling at the beginning of the cut than at steady state.

3 The chip-thickness ratio is small at low cutting speeds and increases as the speed increases. The effect of this on the chip-curl transiency would be that the smaller chip-thickness ratio would give a thicker and stiffer chip, a smaller shear angle, and a longer shear zone. The stiffer chip would give a longer tool-chip contact length. The smaller shear angle would give higher shearing and normal stresses at the shear zone. The longer shear zone coupled with the previous two effects would give a greater stress gradient at the shear zone, a greater wedge angle, and a tighter chip curl. As cutting proceeds, the previous conditions would reverse themselves and result in a looser curl, until at steady state a constant-curvature chip is produced.

4 As soon as cutting starts the temperature at the shear zone goes up by about 350 to 900 F. The increase of the shear-zone temperature would decrease the shear-flow stress and alter the temperature distribution at the shear zone. The effect of decreasing shear-flow stress, as cutting proceeds, would be freer displacement of the shear-zone lamellae and, consequently, a thinner chip would form. In other words, the chip-thickness ratio would be larger and would produce the effects described in item 3.

A change in the temperature distribution at the shear zone, of about 150 to 250 F (Figs. 12 and 13 of the paper), from the state when cutting begins to steady state, would not be sufficient to cause too marked a change in the metal-flow characteristics along the shear zone.[19] It appears, therefore, as if the change in the temperature distribution would not have a very important effect on chip-curl transiency.

5 As soon as cutting starts the temperature at the tool-chip interface increases by about 1300 to 2400 F.[15] Under such high temperatures and extremely high pressures at the tool-chip interface, metallurgical and chemical changes would take place. With metallurgical changes a low shear flow-strength alloy would be formed among the tool and workpiece material components.[16] Simultaneously through the chemical changes, any films adhering to the tool and workpiece surface would react with some of the elements on the surface they come in contact with and form a low shear-strength chemical substance at the tool-chip interface.[17] The combined effects of the previous two results would be to decrease the coefficient of friction as cutting proceeds to a steady state and to produce the results described under item 1. The severe change in friction is evidenced by Figs. 10 and 11 of the paper. Fig. 10 shows very high chip-face deformation (secondary flow) at the beginning of the cut; whereas in Fig. 11 (steady state) the chip face appears to have undergone only "one third" the deformation in Fig. 10.

The sum total of the effects discussed in the preceding five items would give a tighter chip curl or higher curvature at the beginning of the cut, a looser curl as the cut proceeds, and finally a chip of practically constant curvature at steady state.

Fig. 5 shows that the cutting force and the thrust force drop perceptibly ("pip") before the shorter chip is cut. This indicates that there might be a momentary deceleration and relief of pressure on the tool face immediately before the second, shorter cut. These would cause a slight decrease of the tool-face temperature, a contamination of the—now extremely nascent—tool face, and an alteration of the stress distribution at the shear zone. All of these changes would create the same effects discussed in the previous four items. Of course, these changes would not return the workpiece and the tool to exactly the same state as before the longer cut. The state immediately before the second, shorter, cut would be closer to that at the end of the first, longer, cut. For this reason, Fig. 1 of the paper shows that the small chip has a smaller curvature than the long chip, if compared point by point, and has a tendency to reach steady state much sooner.

Author's Closure

The comments of Professors Trigger and Chao and Mr. Kececioglu are most welcome indeed.

Among other things, these discussers have brought up questions bearing on the effect of thermal diffusivity and variations in cutting ratio.

If the curvature, as shown in Figs. 4 or 6, is plotted on semilog paper, it is found to satisfy fairly well the relation

$$(\rho - \rho_e) = ae^{-\alpha x}$$

where

ρ = instantaneous curvature
ρ_e = steady-state curvature
a, α = intercept and slope constants
x = distance along workpiece

The arbitrary constants a and α were found in this way for several toolwork combinations and are shown plotted against thermal diffusivity of the work material in Fig. 16, herewith. It will be seen that generally both constants fall with increasing diffusivity although the scatter is quite large.

A detailed study also was made of the cutting ratio in an effort to measure its variation during the transient. Gage lines were marked on the workpiece before cutting, with approximately $^1/_4$-in. spacing. Their actual spacing was measured carefully, before and after cutting, with a suitable microscope. Fig. 17 of this closure shows the "running" cutting ratio superimposed on the curvature variation, both being plotted against distance traveled. The fluctuations in cutting ratio are about 5 times greater than the maximum possible measurement error. Generally the cutting ratio is substantially constant during the curvature transient.

In reply to some of Mr. Kececioglu's comments it should be pointed out that the difference in appearance of the A and B chips in Fig. 1 of the paper is due to different cutting ratios; and that when the curvature is plotted against distance traveled along the workpiece, the cutting ratio is not involved.

In connection with the influence of the dashpot upon the re-

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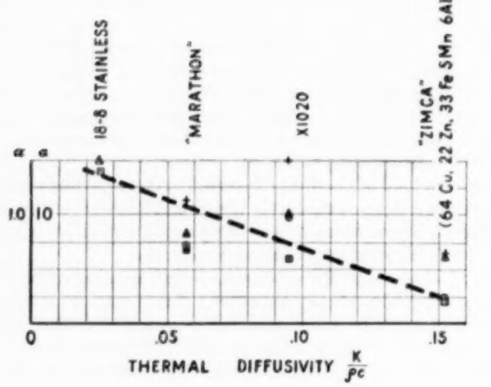
[18] "Carbide High-Velocity Turning," by L. Fersing, Trans. ASME, vol. 73, 1951, pp. 359–374.

[19] "The Effect of Speed and Feed on the Mechanics of Metal Cutting," by B. T. Chao and G. H. Bisacre, Advance Copy, The Institution of Mechanical Engineers, London, England, 1951.



⊙, α for 831 Carboloy tool
⊡, α for K5H Kennametal tool
+, a for 831 Carboloy tool
Δ, a for K5H Kennametal tool

FIG. 16 VARIATION OF SLOPE AND INTERCEPT PARAMETERS α AND a WITH THERMAL DIFFUSIVITY OF WORKPIECE, WHERE
$(\rho - \rho_r) = ae^{-\alpha x}$

Cutting conditions: 0.03 in. depth of cut, 600 fpm, 0 deg rake, 12 deg clear

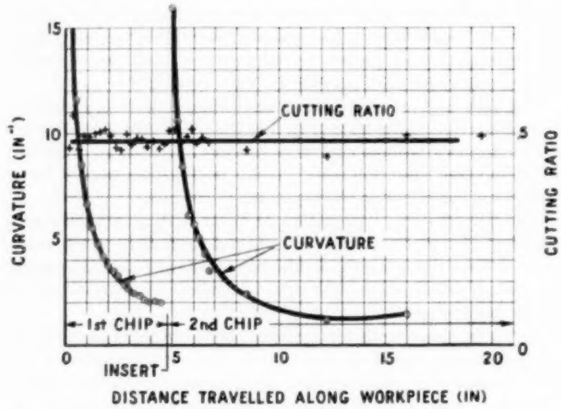


FIG. 17 VARIATION OF CHIP CURVATURE WITH DISTANCE TRAVELED SHOWING SIMULTANEOUSLY CUTTING RATIO

Cutting conditions: 0.003 in. depth of cut, 600 fpm, 0 deg rake, 12 deg clear
Work material: "Marathon" steel (2.22 C, 0.3 Mn, 0.34 S, 0.8 Mo, 12 Cr, 0.2 Va)
Tool material: Kennametal K5H

sponse time of the tool-support system it may be observed that the release of the tool as shown in Fig. 5 of the paper is quite rapid, the dashpot providing something less than critical damping, and so the system may be expected to respond within several milliseconds.

The author is not in full agreement with Professors Trigger and Chao's complaint about the moving heat-source theory. The author does not believe he has neglected the fact "that only a fraction of the energy liberated at the shear zone due to main chip shear goes into the workpiece by conduction." The author's views on these points will be found in a previous discussion.[20]

[20] "The Significance of the Thermal Number in Metal Machining," by B. T. Chao and K. J. Trigger, Trans. ASME, vol. 75, 1953, discussion by R. S. Hahn, p. 115.

Deformation Work Absorbed by the Workpiece During Metal Cutting

By E. G. THOMSEN,[1] J. T. LAPSLEY, JR.,[2] and R. C. GRASSI,[3] BERKELEY, CALIF.

The deformation work absorbed by the workpiece during metal cutting is analyzed. It was shown from photomicrographs that this is appreciable and may constitute the major energy absorbed in metal cutting when the depth of cut becomes small. The so-called "size effect in metal cutting," in part, may be explained by this workpiece deformation. Correcting cutting forces for workpiece deformation in no way detracts from Merchant's analysis, based on the minimum-energy principle. However, the criteria of plasticity in metal cutting and the correlation of metal-cutting data with tensile data remain in doubt.

Nomenclature

The following nomenclature is used in the paper:

τ_s = shear stress during machining in work surface, psi
l_0 = length of surface machined (= initial chip length), in.
l = final chip length, in.
w = width of chip removed, in.
ds, Δs = infinitesimal and incremental surface motion on workpiece, in.
F_C, F_T = nominal cutting force and nominal thrust force, respectively, lb
F_C', F_T' = cutting force and normal force, respectively, causing workpiece deformation, lb
F_C'', F_T'' = net cutting force and net normal force, respectively, available for chip deformation, lb
t_0 = initial chip thickness or depth of cut in orthogonal metal cutting, in.
t = final chip thickness after removal from workpiece, in.
ϕ = shear angle, deg
τ = friction angle, deg
α = back rake angle, deg

Introduction

In metal cutting the waste metal is removed from the workpiece in the form of chips. In the process both the chip and the top layer of the work surface are deformed plastically, the severity of deformation depending on the cutting condition. In the analysis of metal-cutting data, however, the workpiece deformation usually is ignored and all deformation work is considered as going into the chip. The question to be answered then is, how much is this work of deformation of the workpiece and will appreciable error result in analyzing metal-cutting data if all energy of cutting expended is charged to the chip?

Merchant (1)[4] in his basic analysis of the metal-cutting process has concluded that the work of generating a new surface on the workpiece without deforming this surface requires a negligible expenditure of energy. He also concludes from observations of deformation patterns on the workpiece surface as revealed by photomicrographs that the energy of plastic deformation must be small and can be neglected.

There is strong evidence, however, that the workpiece deformation may not be entirely negligible as revealed by the results of various investigations on determinations of residual stress and depth of cold work in the workpiece reported in the literature (2, 3). It has been shown, for example, that grinding mild steel to a depth of 0.0003 in. caused a maximum residual stress of 75,000 psi and a depth of cold work of approximately 0.012 in.

It is the purpose of this paper to show evidence that the work of deformation of the workpiece in orthogonal cutting is not negligible, especially for small chips, and that proper accounting should be made in analyzing the forces in cutting.

Experimental Results

Figs. 1 to 5, inclusive, show photomicrographs of machined surfaces of SAE 1113 steel. The machining was performed in a hydraulic shaper and the workpieces were cut from cold-drawn bar stock in such a way that the banding of the nonmetallic inclusions was perpendicular to the machined surface. This orientation of the nonmetallic inclusions facilitated visualization of surface displacement as evidenced by distortion at the surface in the direction of tool motion. It may be seen that for relatively light cuts the surface displacement and the depth to which the material in the workpiece appears to be affected is smaller than for heavy cuts. Severe penetration of cold work and surface displacement are obtained when a built-up edge is formed on the cutting tool as shown in Figs. 4 and 5.

It was not possible to give exact quantitative data of surface displacement from micrographic studies of machined surfaces. This was due to the tendency of the bands of nonmetallic inclusions to approach tangency to the surface and the surface of the workpiece to become slightly rounded during polishing. An estimate of the surface displacement of a number of machined surfaces examined, ranging in depth of cut from 0.001 to 0.008 in., is 0.003 to 0.005 in., the smaller displacement being obtained with light cuts.

It was observed in an earlier investigation published by the authors (4) that the cutting force and thrust force for orthogonal cutting (end-cutting in a lathe) of Shelby tubing were approximately linear functions of depth of cut for several rake angles. These data are reproduced in Fig. 6. Further investigations were made on other materials, under similar cutting conditions, which revealed that this was also the case for these experiments for the range of cuts and several cutting conditions explored. These results are shown in Figs. 7, 8, and 9. The fact that these curves do not pass through the origin is the basis of the argument to be presented that the work of deformation of the workpiece is not necessarily negligible.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, June 26, 1952. Paper No. 52—F-24.

[4] Numbers in parentheses refer to Bibliography at end of paper.

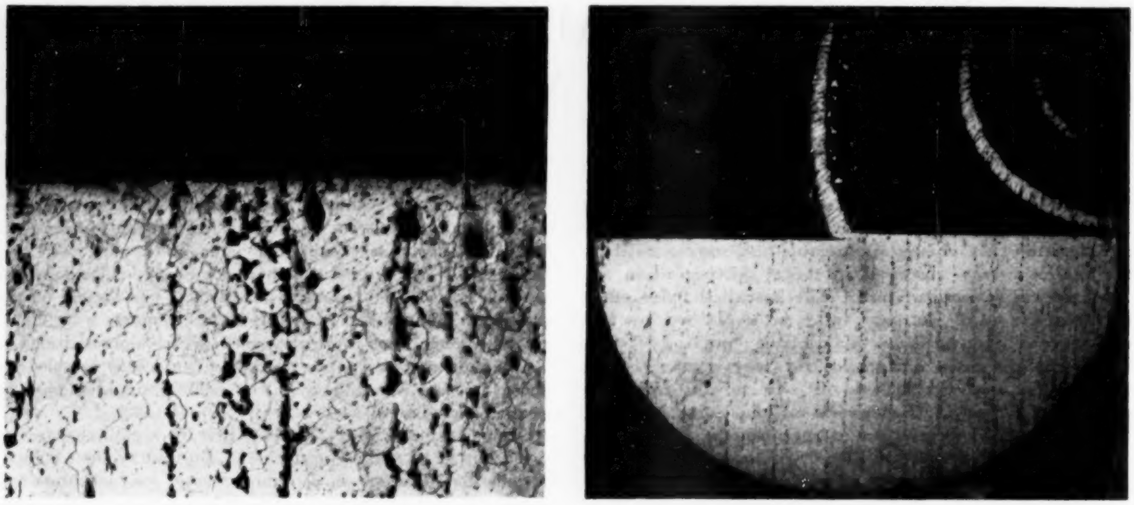

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Fig. 1 Photomicrographs of Chip and Workpiece Deformation of SAE 1113 Steel

(Depth of cut = 0.0018 in., 20-deg back rake, 2-deg end clearance, dry, orthogonal cutting.)

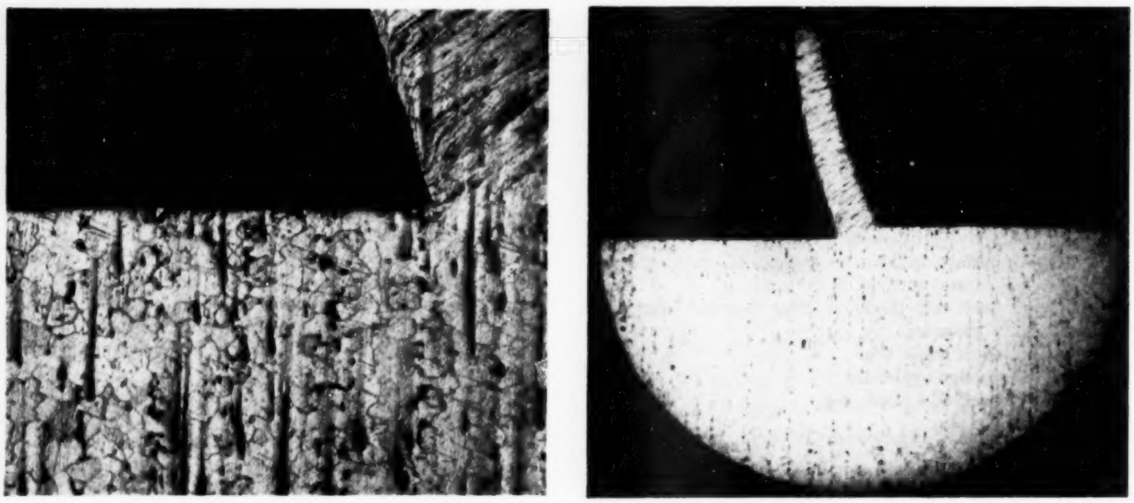

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Fig. 2 Photomicrographs of Chip and Workpiece Deformation of SAE 1113 Steel

(Depth of cut = 0.0044 in., 20-deg back rake, 2-deg end clearance, dry, orthogonal cutting.)

The forces in Figs. 7, 8, and 9 were obtained with the dynamometer shown in Fig. 10 and were recorded in terms of deflection on a two-channel Brush recorder. The design of the tool dynamometer is such that each tool force is independent of the other, thus requiring no corrections. The dynamometer was calibrated with dead weights in position for cutting. Cutting ratios, which are a measure of chip deformation, were obtained from chip-length measurements. This was accomplished by marking the tubular workpiece by a saw cut on its surface parallel to its axis. This saw cut provided markings on the continuous chip; hence its original length between saw cuts was known. The final length was obtained by passing it through the special measuring device shown in Fig. 11, after careful annealing and straightening of the chip.

Discussion

In order to calculate the workpiece-deformation work it is expedient to consider a somewhat analogous deformation process as represented by piercing a hole. Fig. 12 shows two stages of a pierced hole obtained by copper furnace brazing, in a hydrogen atmosphere, two ground edges of mild-steel plate marked with a square network of lines. The plate was then centered in a die set and partially pierced. The distorted network was made visible after separating the two parts of the plate in a furnace.

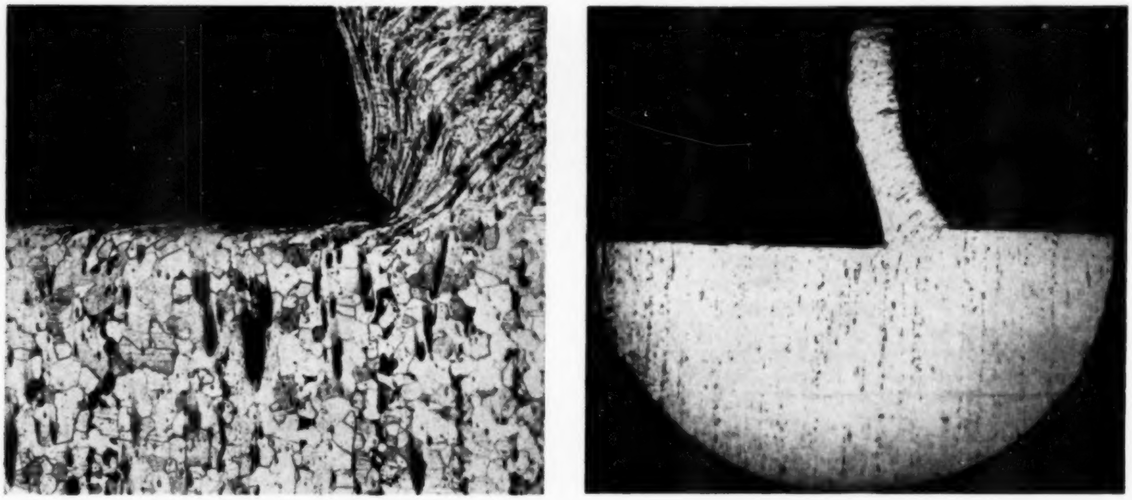

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Fig. 3 Photomicrographs of Chip and Workpiece Deformation of SAE 1113 Steel
(Depth of cut = 0.012 in., 20-deg back rake, 2-deg end clearance, dry, orthogonal cutting.)

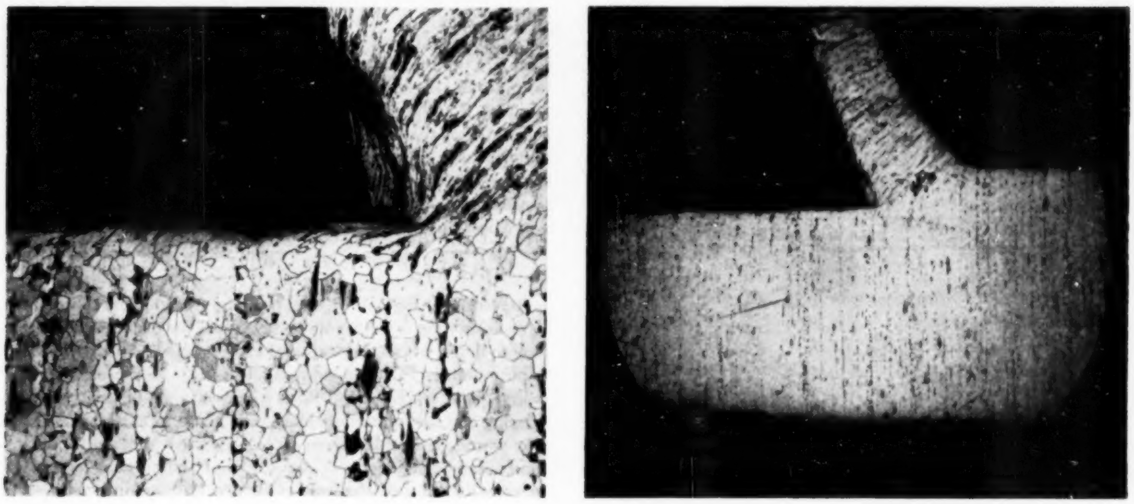

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Fig. 4 Photomicrographs of Chip and Workpiece Deformation of SAE 1113 Steel
(Depth of cut = 0.033 in., 20-deg back rake, 2-deg end clearance, dry, orthogonal cutting.)

It may be noted that fracture has not yet occurred in spite of the fact that the punch penetrated a considerable distance into the plate. The total work absorbed by the plate in this case is given by the average punch force times the distance the punch has penetrated. This deformation work has resulted in severe deformation, principally by shear, in the region near the edges of the die and punch. Let it now be assumed that workpiece deformation in metal cutting can be visualized in a similar manner as shown schematically in Fig. 13. Let it also be assumed that the chip is separated from the workpiece without absorption of energy in the chip and that all work expended goes into the workpiece. Hence a chip displacement of a distance ΔS, Fig. 13, results in the following workpiece-deformation work

$$wk = \int_0^{\Delta s} \tau_s wl ds \qquad [1]$$

If τ_s is the flow stress of the material and for a first approximation is assumed constant, the work becomes

$$wk = \tau_s wl \Delta s \qquad [2]$$

In machining, however, the top layer of the work is removed incrementally in the form of a chip as the tool advances a distance l. If the work absorption of the workpiece is assumed to be the same for the real cutting process and for the analogous process

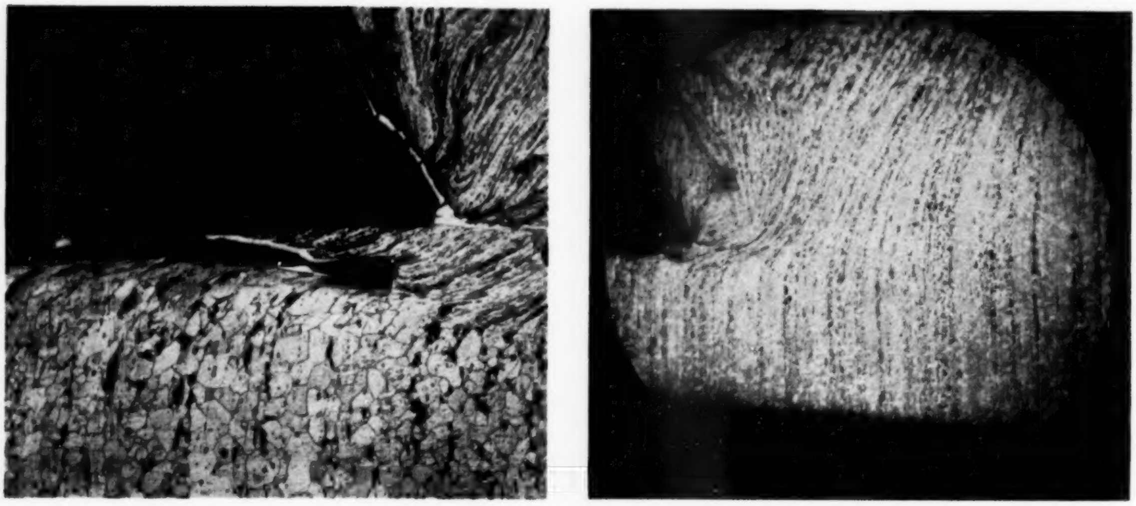

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Fig. 5 Photomicrographs of Chip and Workpiece Deformation of SAE 1113 Steel

(Depth of cut = 0.057 in., 20-deg back rake, 2-deg end clearance, dry, orthogonal cutting.)

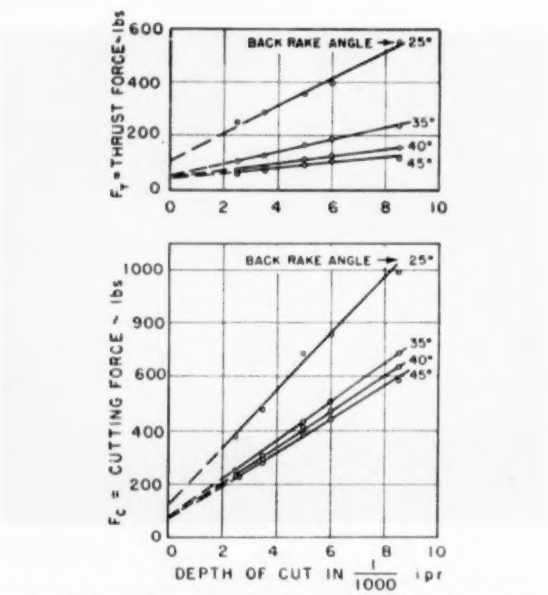

Fig. 6 Cutting Force and Thrust Force—Orthogonal Cutting, Lathe—as Function of Depth of Cut for Several Back-Rake Angles

(Material: 6-in-OD Shelby tubing, cut dry, HSS, 6-deg end clearance, 0.475 in. width of cut, and 90 fpm surface speed.)

shown in Fig. 13, then

$$wk = \tau_s wl\Delta s = (\tau_s w\Delta s)l = F_c''l \qquad [3]$$

where F_c' is the equivalent force responsible for the work done. Subtracting this portion of work from the total work of cutting yields

$$F_c l - F_c'l = F_c''l$$

or

$$F_c - F_c' = F_c'' \qquad [4]$$

where F_c is the total cutting force and F_c'' is the force causing chip deformation. In order to calculate the force for chip deformation F_c'', it is necessary to know the total force F_c, and the force F_c'. F_c can be obtained from a dynamometer, while a first estimate of the magnitude of F_c' can be obtained by calculating $\tau_s w\Delta s$. For the experiments with SAE 1113, using end-cutting, the width of the chip w was $^1/_8$ in. and the yield strength of the material was approximately 72,000 psi. Assuming an average surface motion of 0.004 in. and simple shearing with no work-hardening, then $\tau_s = 72{,}000/\sqrt{3} = 41{,}600$ psi, if based on the von Mises criterion for flow

$$F_c' = \tau_s w\Delta s = 41{,}600 \times 0.125 \times 0.004 = 20.8 \text{ lb} \qquad [5]$$

Referring to Fig. 9, it is seen that the straight line representing F_c as a function of depth of cut intersects the ordinate above zero at zero depth of cut, assuming of course that linear extrapolation is permissible. The intercept in this case is approximately 20 lb and therefore Equation [5] is a fair estimate of this value considering the assumptions which were made in calculating it. If it is assumed that the force F_c' represented by this intercept is constant for any depth of cut and that the cutting force-depth curve is linear, then the force of removing the chip is reduced by this amount, and is given by F_c'', hence

$$\frac{F_c''}{t_0} = \frac{\text{Force for chip deformation}}{\text{Depth of cut}} = \text{const} \ldots\ldots \qquad [6]$$

If Equation [6] represents a true accounting of the cutting force available for chip deformation then the "size effect in metal cutting" found by Backer, Marshall, and Shaw (5) in recent experiments with micromilling and grinding SAE 1112 steel is nonexistent. It was shown by these investigators when applying force values obtained from dynamometer readings without correction for workpiece deformation that the calculated shear stresses on the shear plane were phenomenally high. They found for example, that the shear stress to cause plastic flow in chips for micromilling was of the order of 2 to 3 times, and for grinding nearly 20 times as large as those found with turning. These authors explained this anomaly in terms of the size ef-

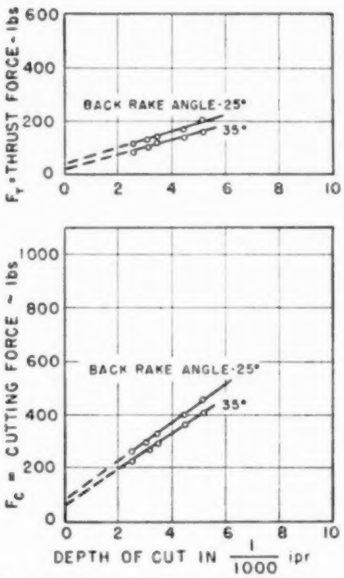

Fig. 7 Cutting Force and Thrust Force—Orthogonal Cutting, Lathe—as Function of Depth of Cut for 25-Deg and 35-Deg Back-Rake Angles

(Material: 5.800-in-OD SAE 4130 steel tubing, Cimcool coolant, HSS, 10-deg end clearance, 0.300 in. width of cut, and 42 fpm surface speed.)

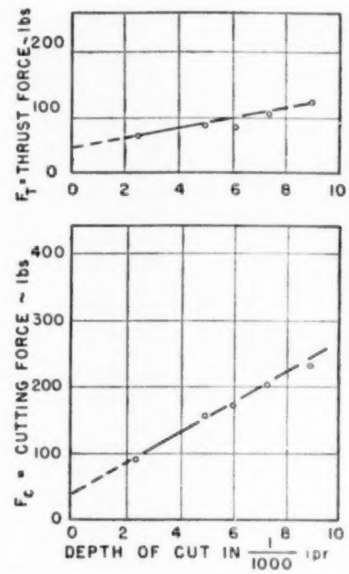

Fig. 8 Cutting Force and Thrust Force—Orthogonal Cutting, Lathe—as Function of Depth of Cut for 30-Deg Back-Rake Angle

(Material: 4.260-in-OD, 61S-O aluminum alloy tubing, kerosene, HSS, 10-deg end clearance, 0.300 in. width of cut, and 114 fpm surface speed.)

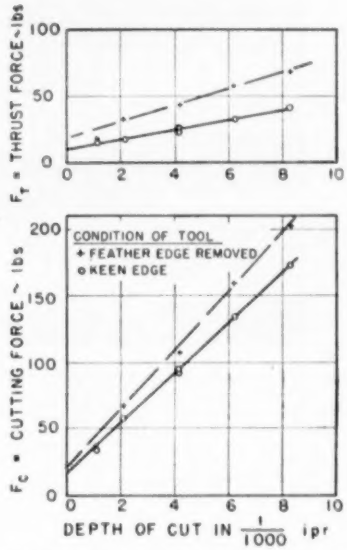

Fig. 9 Cutting Force and Thrust Force—Orthogonal Cutting, Lathe—as Function of Depth of Cut for Two Conditions of Cutting Edge

(Material: $2^3/_8$-in-OD SAE 1113 tubing machined from a drawn bar, cut dry, HSS, 20-deg back-rake, 8-deg end clearance, 0.125-in. width of chip, and 100 fpm surface speed.)

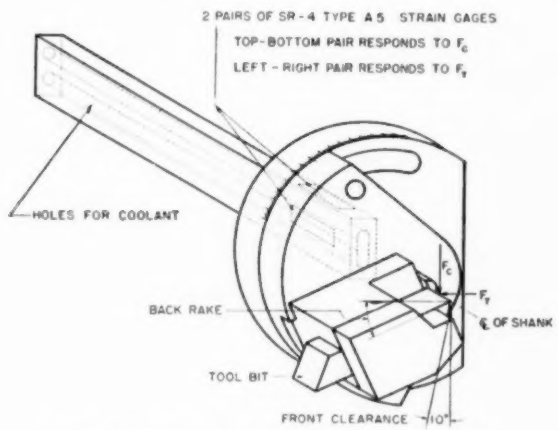

Fig. 10 Schematic Drawing of Tool Dynamometer

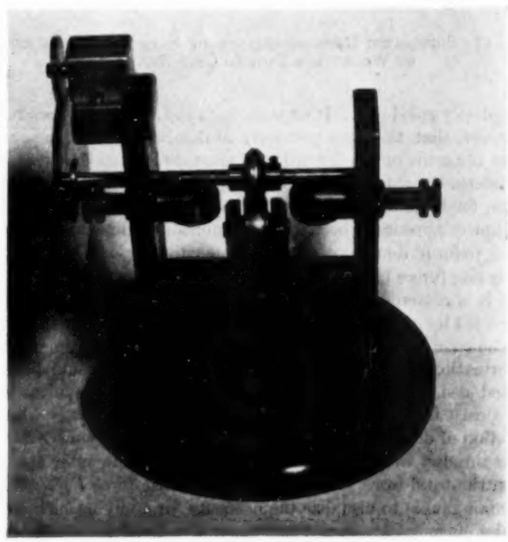

Fig. 11 Device for Continuous Chip Length Measurement

fect observed in testing metals by conventional means, as, for example, in the tensile test, where the strength of the material appears to increase as the size of the specimen decreases. Inasmuch as portions of the chips in micromilling and grinding are extremely thin as compared to turning, these authors were justified in reaching their conclusions in attributing the large shear stresses required for removing small chips to the so-called size effect. If, however, Equation [6] is a statement of fact then it is readily apparent that the ratio of energy absorption of workpiece/chip, and hence the ratio F_c'/F_c'' increases as the depth of cut decreases, approaching infinity as the depth of cut approaches zero and the "effect of size" need not be postulated.

Returning to the photomicrographs presented in Figs. 1 to 5, inclusive, it is evident, however, that Equation [6] does not represent the true distribution of forces in metal cutting. The surface displacement, as well as the depth of the material in the workpiece affected by the chip removal, increases with increasing depth of cut. Thus, assuming that a constant portion of the total energy of cutting is absorbed by the workpiece is incorrect and the size effect in metal cutting previously referred to is not

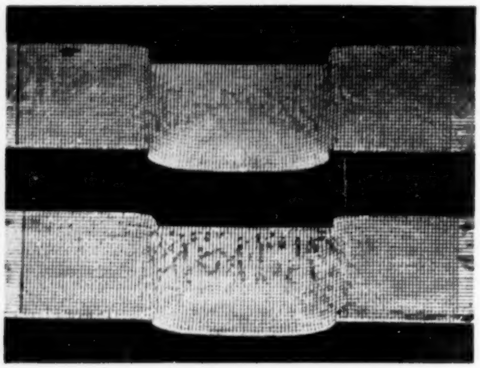

Fig. 12 Flow Pattern During Piercing a 0.475-In-Diam Hole in Low-Carbon Steel Plate

(Flow pattern is made visible by marking a square grid on edges of two plates subsequently copper-brazed before piercing. Experimental work and photographs by Dr. C. T. Yang.)

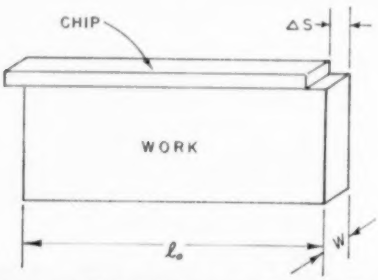

Fig. 13 Schematic Representation of Surface Displacement of Workpiece During Chip Removal

completely ruled out. It appears from the arguments presented, however, that the force necessary to deform the workpiece may be of the same or greater order of magnitude than that required to deform the chip when the depth of cut is small. On the other hand, for large depths of cut the energy required to deform the workpiece appears to become less significant and neglect of workpiece deformation may not introduce too serious an error in analyzing forces in metal cutting.

If it is reasonable to assume that the cutting force F_c can be corrected by a constant value F_c' as represented by the intercept on the cutting force-depth curve to obtain the approximate chip deformation force F_c'', then the thrust force F_T should be corrected also. Referring to Figs. 6 to 9, inclusive, it is evident that the thrust force F_T for the various cutting conditions is a linear function of depth of cut and that the experimental curves do not pass through the origin. Hence, subtracting the intercept F_T' from the total force yields a corrected thrust force F_T''. While it is not simple to visualize the necessity for applying such a correction, it may be assumed that the underside of the cutting edge is bearing on the machined surface and that the force F_T' is contributing nothing to chip deformation. If such a correction is permissible it is at once evident that the deformation of the workpiece surface is more complicated than assumed previously, because it takes place under the action of two mutually perpendicular forces.

In the light of what has been presented it is of interest to examine theories on metal cutting when the cutting forces have been corrected for workpiece deformation. Merchant (6) has derived the expression

$$2\phi + \tau - \alpha = 90 \text{ deg} \qquad [7]$$

by applying the principle of minimum energy to metal cutting. Hill (7) has questioned the validity of applying this principle to the metal-cutting process from a purely theoretical point of view but shows no data to substantiate his claim. Merchant (8) further introduced a plasticity condition to account for the fact that his experimental data did not fit the angular condition predicted by Equation [7]. He found that by introducing a linear functional relationship between the shear stress on the shear plane and the normal stress on this same plane, better correlation between experiment and theory was achieved. The authors investigated the plasticity (9) condition by comparing chip deformation in metal cutting with deformation in a tensile test bar of the same metal. Good correlation was achieved with Shelby tubing under various cutting conditions. Backer, Marshall, and Shaw (10) found that similar good agreement was obtained when turning data were compared with tensile deformation data of SAE 1112 steel.

Figs. 14 and 15 show graphically the minimum-energy principle represented by the solid lines marked ($2\phi + \tau - \alpha = 90$ deg). The points plotted on these graphs represent the reduced experimental data corrected for workpiece deformation as given by the intercepts of the force-depth curves discussed previously. These experimental data and calculated data are given in Table 1.

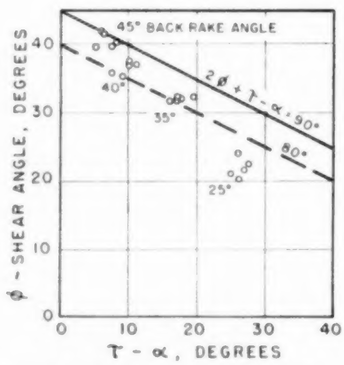

Fig. 14 Comparison of Experimental Metal-Cutting Results With Minimum-Energy Principle Given by $2\phi + \tau - \alpha = 90$ Deg

(Material: Shelby tubing for various rake angles and feeds. Data from Table 1.)

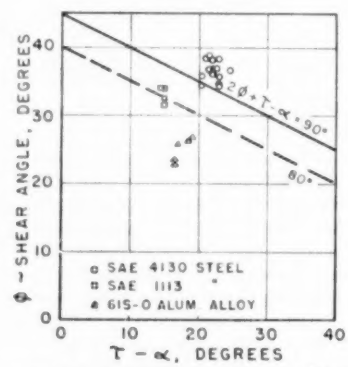

Fig. 15 Comparison of Experimental Metal-Cutting Results With Minimum-Energy Principle Given by $2\phi + \tau - \alpha = 90$ Deg for Various Metals and Cutting Condition

(Data from Table 1.)

TABLE 1 REDUCED DATA ORTHOGONAL METAL CUTTING IN LATHE

Test description	Back-rake angle α, deg	Depth of cut, in.	Cutting ratio $\frac{r_0}{r} = \frac{l}{l_0}$	Cutting force components — Uncorrected — Cutting F_C, lb	Thrust F_T, lb	Corrected — Cutting F_C'', lb	Thrust F_T'', lb	Shear angle ϕ, deg	Coefficient of friction, $\mu = \left(\frac{F_T'' + F_C'' \tan\alpha}{F_C'' - F_T'' \tan\alpha}\right)$	$\tau - \alpha$, deg
Mat'l—Shelby tubing 6 in. OD; 0.475 in. chip width, HSS; 6 deg end clearance; V = 90 fpm; dry cutting	25	0.0025	0.358	380	224	260	124	20.9	1.21	25
		0.0035	0.366	475	281	355	181	21.5	1.28	27
		0.005	0.407	643	357	523	257	24.0	1.24	26
		0.006	0.345	728	398	608	298	20.1	1.24	26
		0.0085	0.383	992	551	872	451	22.4	1.30	27.5
	35	0.0025	0.527	254	102	184	52	31.6	1.23	16
		0.0035	0.528	306	122	236	72	31.9	1.28	17
		0.005	0.529	433	166	363	116	32.0	1.31	17.5
		0.006	0.533	507	184	437	134	32.2	1.40	19.5
		0.0085	0.532	675	234	605	184	32.0	1.28	17
	40	0.0025	0.585	232	71	162	21	35.7	1.09	7.5
		0.0035	0.580	296	87	226	37	35.4	1.16	9
		0.005	0.611	412	112	342	62	37.5	1.20	10
		0.006	0.606	475	127	405	77	37.2	1.23	11
		0.0085	0.606	634	153	564	103	37.2	1.20	10
	45	0.0025	0.670	232	68	162	18	41.9	1.25	6
		0.0035	0.670	285	77	215	27	41.9	1.29	6
		0.005	0.649	386	94	316	44	40.2	1.32	8
		0.006	0.642	443	102	373	52	39.6	1.01	5
		0.0085	0.646	581	117	511	67	39.9	1.30	7.5
Mat'l—SAE 4130 5.8 in. OD; 0.300 in. chip width, HSS; 10 deg end clearance; V = 41.7 fpm; "Cimcool"	25	0.00256	0.573	258	114	178	74	34.4	1.10	23
		0.00315	0.573	298	131	218	81	34.4	1.01	20.5
		0.00343	0.567	331	145	251	105	34.0	1.10	23
		0.00442	0.594	403	174	323	134	35.7	1.10	23
		0.00517	0.599	457	204	377	154	36.0	1.08	22
	30	0.00256	0.599	254	113	184	83	36.5	1.39	24.5
		0.00315	0.602	276	114	206	84	36.7	1.29	22
		0.00343	0.605	312	132	242	102	36.9	1.31	22.5
		0.00442	0.625	370	150	300	120	38.2	1.27	22
		0.00517	0.628	417	164	347	134	38.4	1.24	21
	35	0.00256	0.573	228	85	168	65	35.7	1.49	20.5
		0.00315	0.599	267	105	207	85	36.7	1.51	21.5
		0.00343	0.619	298	121	238	101	38.1	1.60	23
		0.00442	0.625	364	140	304	120	38.6	1.51	21.5
		0.00517	0.630	406	161	346	141	38.9	1.55	22
Mat'l—61S-O Alum. 4.5 in. OD; 0.300 in. chip width, HSS; 10 deg end clearance, V = 114 fpm; kerosene	30	0.00245	0.402	91	55	51	15	23.5	1.05	16.5
		0.00490	0.450	157	81	117	41	26.7	1.16	19
		0.00610	0.467	174.5	79.5	135	40	22.8	1.05	16.5
		0.00730	0.435	204	91	164	51	25.7	1.08	17
		0.00390	0.443	240	108	200	68	26.3	1.14	18.5
Mat'l—SAE 1113 2.25 in. OD; 0.125 in. chip width, HSS; 8-deg end clearance; V = 114 fpm; dry cutting	20[a]	0.00107	...	38	18	...	...	...	...	...
		0.00205	0.578	68	32.5	48	12.5	34.0	0.69	14.5
		0.00412	0.577	108	43.5	88	23.5	34.0	0.70	15
		0.00620	0.555	160	58	140	38	32.5	0.70	15
		0.00824	0.536	202	69	182	49	31.5	0.70	15
	20[b]	0.00107		34	16.5					
		0.00205		56	16.5					
		0.00205		56	16.5					
		0.00412		96	25.5					
		0.00412		93	23.5					
		0.00412		93	24.5					
		0.00620		135	32.5					
		0.00824		173	42					

[a] Feather-edge removed from tool.
[b] Keen cutting edged tool.

Referring to Fig. 14, it may be seen that the experimental points for Shelby tubing lie between $2\phi + \tau - \alpha = 90$ and 80 deg, except for the data obtained with a 25-deg back-rake angle. It appears from an interpretation of these data that for large rake angles the minimum-energy principle is satisfied, but that for small rake angles the experimental points depart considerably from the solid line. If the minimum-energy principle is accepted as a statement of fact, then a possible explanation of the departure of the experimental points from the solid curve may be that a built-up edge was present on the tool face in varying degrees, becoming more severe as the rake angles decreased. It is of interest to note that the experimental data uncorrected for workpiece deformation plots substantially the same as the corrected values given in Fig. 14.

Fig. 15 shows a similar comparison for SAE 4130 steel, SAE 1113 steel, and aluminum alloy 61S-O under varying cutting conditions. The SAE 4130 data fall above the solid line, while the data for SAE 1113 and aluminum alloy 61S-O fall below this line. Cutting for alloy 61S-O was definitely with a built-up edge and hence the departure of these points from the theoretical curve is of no significance. The departure above and below this line for the other alloys is not clear, however, and naturally leads to the conclusion that the cutting forces and thrust forces were not corrected properly or the minimum-energy principle is not completely applicable.

Examining the plasticity conditions as formulated by Merchant, and referred to earlier, it is not clear if sufficient data are available to come to any conclusion. The essence of Merchant's condition is given by a straight line parallel to, but below the solid lines marked 90 deg in Figs. 14 and 15. It is evident that there is no significant trend for any such line. It is also of interest to note that correlation between the metal-cutting data and data of a static tensile test, as advocated by the authors, becomes poorer when corrections for workpiece deformation are made. The data for Shelby tubing examined on this basis revealed that the metal-cutting data fell consistently below the extrapolated true stress-true strain curve of the reduced tensile-test data. It appears from this that the acceptance of correction of tool forces for workpiece deformation beclouds the issue of what role the plastic properties of the work material play in the metal-cutting process.

Conclusions

1 It was shown that the work of deformation absorbed by the workpiece may be a major portion of the total cutting work when the chips are small.

2 For several materials examined it appears that the intercept of the cutting force at zero depth of cut is an approximate estimate of the force required to deform the workpiece and hence is not available for chip deformation.

3 The workpiece deformation appears in part to account for the size effect encountered in the removal of small chips. It

was pointed out, however, that this effect may still be present but to a smaller extent than believed heretofore.

4 Theories of metal cutting were examined by arbitrarily correcting tool forces by subtracting the force intercepts obtained from extrapolating the force-depth curves linearly to zero depth of cut. The following tentative conclusions were reached:

(*a*) The principle of minimum energy for determining the shear angle appears to be operative.

(*b*) The plasticity condition in metal cutting, based on the hypothesis that the shear stress on the shear plane to cause plastic deformation is a linear function of the normal stress on this same plane, could not be verified since the corrected experimental points showed no significant trend.

(*c*) The correlation between tensile stress-strain relationships and metal-cutting data yielded poorer correlation when the tool forces were corrected as compared with uncorrected forces.

ACKNOWLEDGMENT

The authors wish to acknowledge the contribution of M. Silverman and E. Robitschek, former graduate students in Mechanical Engineering at the University of California, for the design of the tool dynamometer and chip-measuring device as well as the cutting data for SAE 4130 and aluminum alloy 61S-O. Further grateful acknowledgment is given to Professor Yang, now of M.I.T., who obtained the flow patterns during metal piercing, and to Mr. C. Lockrem of the University of California production laboratories for his excellent assistance in setting up experimental equipment and taking experimental data.

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8 Reference (6), p. 322.

9 Reference (4), p. 983.

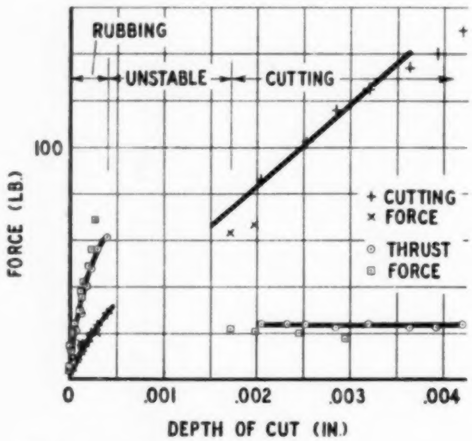
10 Reference (5), p. 6.

Discussion

R. S. HAHN.[5] The authors have presented an interesting paper. They have considered the total energy of cutting $F_c l$, to be divided between workpiece deformation and chip deformation. Is it not possible for a certain amount of energy to be dissipated in the form of heat at the tool-chip interface without producing deformation in the chip? Then only a part of F_c'' would contribute to chip deformation. In other words, would not the speed of cutting affect Equation [6] of the paper?

It might be interesting to show the behavior of the machining forces in the region of very small depths of cut. Fig. 16 of this discussion shows the cutting and thrust forces beginning with depth of cuts of 0.000020 in. In the range from 0 to 0.0005 in.

[5] Consulting Engineer, The Heald Machine Company, Worcester, Mass. Mem. ASME.



Material: Leaded bronze
Tool: Carboloy 883; 0 deg rake; 0.458 deg clearance
Speed: 20 fpm
Width of cut: $^1/_4$ in.
Highly work-hardened surface

FIG. 16 FORCE VERSUS DEPTH OF CUT

the slope of the cutting-force curve is greater than in the larger range. In this range the metal removed is in the form of fine dust. The surface being machined here is highly work-hardened from previous machining, and being harder, would be expected to produce a steep slope to the cutting-force curve.

It would seem desirable to consider the effect of the increased hardness of the surface layers, especially at small depths of cut in connection with the authors' theory.

E. K. HENRIKSEN.[6] The key to the ultimate understanding of the cutting process is the behavior of the material directly in front of the cutting edge. During cutting this place is almost inaccessible for observation and measurement, and so we have to try to approach it from the two different sides, through the chip, and through the workpiece.

The aim of the present paper is to provide a more correct analysis of the cutting forces by drawing some conclusions from the deformed layer. It is thought that the authors have been successful in their attempt, and also that they are opening up a new road by which it is possible now to go even a few steps further than the authors have done.

The authors have found a linear relation between the cutting force F and the depth of cut which for this case of orthogonal cutting is equivalent to area of cut, and they discuss the possibility of a "size effect."

The existence of a "formal size effect" has been known for a long time. Some years ago Prof. H. Friedrich in Chemnitz suggested the equation

$$F = c_1 + c_2 a \qquad [8]$$

where a is area of cut. A former colleague of the writer's in Copenhagen, Prof. E. Thaulow[7] formulated a linear relation in a different manner by introducing the effective length l of the cutting edge, thus

$$F = c_1 a + c_2 l \qquad [9]$$

[6] Professor in Charge, Department of Materials Processing, Cornell University, Ithaca, N. Y.

[7] "Maskinarbejde," by E. Thaulow, in co-operation with E. K. Henriksen, Copenhagen, second edition, vol. 1, 1942, p. 246.

For orthogonal cutting with a straight cutting edge, Equations [8], [9] are identical and they are also formally identical with the authors' graphs shown in Figs. 6 to 9, because a is proportional to l; but for other cases, where a and l are independent variables, Thaulow's Equation [9], herewith, is a step beyond Friedrich's Equation [8], because it expresses the viewpoint that there are two contributions to the total cutting force, one (c_1a) from the area of cut, and the other one (c_2l) from the cutting edge, the edge contribution being physically and mathematically independent of the area contribution. Here is actually formulated a "physical size effect," probably for the first time, because this equation was given in machine-tool lectures in the 1920's, and published in 1926, in the first edition of the book cited.[7]

The intercept F_c' in Figs. 6 to 9 of the paper is the same as c_2l in Equation [9], herewith.

In the future probably we will have to deal with several classes of size effect. Be it therefore suggested that the particular size effect discussed by Backer, Marshall, and Shaw (authors' Bibliography 5) be termed "molecular size effect."

Ignoring any possible effect of depth of cut the authors consider F_c', defined as the intercept on the ordinate axis, as the (constant) force responsible for the work deformation, and find a good agreement between this value of the deformation force and the value determined by the physical properties of the material.

Then, they recognize the fact that heavier cuts produce more work deformation and, consequently, require values of F_c' which are higher than the one found as the intercept; but the method described, by its very nature, does not permit determination of any other value of F_c' than the intercept, which actually is the hypothetical value of F_c' for zero depth of cut.

Now it will be shown that it is possible to get some information about the variation of F_c' with varying depths of cut.

There must exist a relation, on the one side, between the depth of cut, and, on the other side, (a) the size of the displacement in the surface, (b) the depth of the deformed layer. To these two factors can be added (c) the resultant of all the residual stresses in the surface layer. Graphs and data for this relation are given in the writer's previous publications.[8]

There also must be a relation between the deformation force F_c' (the true value, not the intercept) and the resultant (call it R) of the residual stresses.

Although this relation has not yet been determined in all detail the following can be said:

1 F_c' and R are not identical.
2 F_c' and R are of the same order of magnitude.
3 F_c' is somewhat larger than R.

The writer has no data on SAE 1113 steel, as used by the authors. The closest approximation to it is, in all probability, a 0.10 per cent carbon steel, annealed for stress-relief; data for the resultant residual stress in this material are given in the graph, Fig. 17 of this discussion. In order to compare with the authors' values for SAE 1113, as given in their Fig. 9, the resultants R have been calculated for a workpiece of $^1/_8$ in. width, and for a tool with 20-deg back-rake angle. The results are shown in Fig. 18, herewith.

Several observations can be made from this curve: (1) there is a considerable size effect[9] on R, (2) the intercept on the ordinate

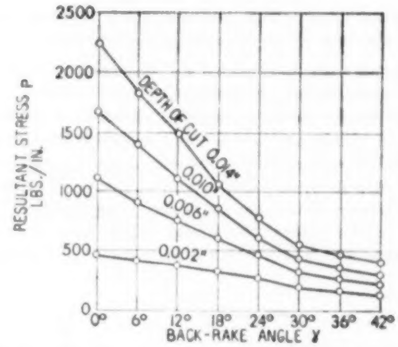


FIG. 17 VARIATION OF RESULTANT STRESS p LB PER IN. WIDTH OF SURFACE, IN RELATION TO DEPTH OF CUT AND BACK-RAKE ANGLE OF TOOL, IN ORTHOGONAL CUTTING
(Material 0.10 carbon steel.)

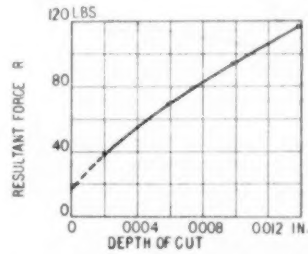


FIG. 18 VARIATION IN RESULTANT FORCE R IN $^1/_8$-IN-WIDE WORKPIECE IN RELATION TO DEPTH OF CUT
(Orthogonal cutting; 20 deg back-rake angle on tool; material 0.10 carbon steel.)

is 17 lb, which should be compared with the authors' value of approximately 20 lb for F_c', (3) R increases rapidly with depth of cut.

Therefore F_c' may very well increase so fast with increasing depth of cut that it cannot be totally ignored in accurate metal-cutting-force analysis; fortunately it is possible to get some idea about its size.

No conclusion shall be attempted at this time as to the molecular size effect by Backer, Marshall, and Shaw; its existence does not necessarily seem to be ruled out.

Now, where does F_c' occur?

The authors conclude that there *may* be a force F_T' between the machined surface and the relief side of the tool, at and near the cutting edge.

The writer feels that such a force *must* exist, and that it is not insignificant in size. There is considerable evidence for the existence of such a force:

1 The present paper provides one such piece of evidence.

2 When the authors, in 1949, presented their paper on "Correlation of Plastic Deformation During Metal Cutting With Tensile Properties of Work Material" the writer was in the position to show how the application of a set of forces (F_T', etc.) would provide an improved agreement between observation and theory in the authors' paper. The method of analysis used in the determination of F_T' has not yet been published.

3 No edge is mathematically sharp; any edge is rounded; it is only a matter of magnification if you want to see it. Compare Fig. 19 which is drawn from an illustration by Fischer.[10]

[8] "Zerspanung und Eigenspannungen," by E. K. Henriksen, Ingeniorvidenskabelige Skrifter, A, No. 43, Copenhagen, Denmark, 1937.

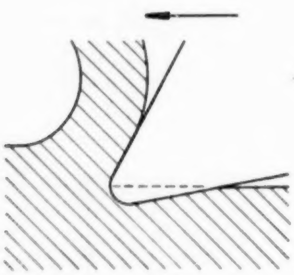
"Residual Stresses in Machined Surfaces," by E. K. Henriksen, Transactions of the Danish Academy of Technical Sciences, No. 7, Copenhagen, Denmark, 1948.

[9] Size effect on residual machining stresses is also evident from the curves in figs. 16 to 41 in the writer's "Residual Stresses in Machined Surfaces."[8]

[10] "Die Werkzeugmaschinen," by Hermann Fischer, Berlin, Germany, 1900, p. 9, fig. 1.

4 There are more considerations which substantiate the existence of the force F_T'; they will be omitted here.

The bearing area upon which F_T' is acting, is small, and the surface is clean metal; consequently there must be a high coefficient of friction and a frictional component of a considerable size, Fig. 20 of this discussion. This frictional force may constitute the whole of the force F_e' which drags the surface and deforms the surface layer. It is, however, more likely that the frictional force supplies only a part of the "dragging force" F_e'; the balance must be made up by stresses which have their origin on the tool face and which proceed in more or less curved paths



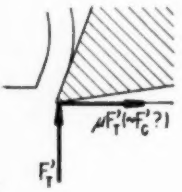


FIG. 19 (*left above*) ACTION OF CUTTING TOOL WITH ROUNDED EDGE (After Hermann Fischer.)

FIG. 20 (*right above*) FORCE COMPONENTS ON UNDERSIDE OF CUTTING TOOL

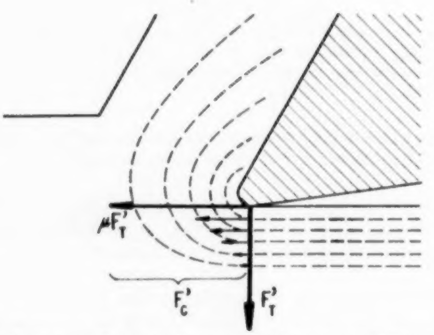


FIG. 21 SYMBOLIC REPRESENTATION OF FORCES AND STRESS FLOW, WHICH PRODUCE DEFORMED LAYER AND RESIDUAL STRESSES IN AND BELOW MACHINED SURFACE

which penetrate the shear zone, and terminate in the deformed layer, as shown symbolically in Fig. 21, herewith. The structures found in chips, particularly when a built-up edge is present, are indications in the same direction.[11]

When the exact analysis of the stress system is completed (Fig. 21 is symbolic only), then we have made a great step toward the realistic answer to the problem of plasticity in cutting, and we may expect to find that the agreement between the authors' observations and the final theory is actually much better than they want to state today. This means again that "chip deformation" is closely related to "surface-layer deformation."

K. J. TRIGGER[12] AND B. T. CHAO.[13] The authors have presented a most interesting analysis of the deformation energy absorbed by the workpiece during chip formation. They have indicated the desirability of considering such work in the interpretation of metal-cutting data, particularly at light feeds and/or depths of cut. The writers commend them for the fine piece of work which has been carried out in the interest of metal-cutting science.

Analysis of metal-cutting data under conditions in which a type 2 chip is formed reveals that a decrease in feed (i.e., "depth of cut" in orthogonal cutting, Figs. 6 to 9 of the paper) with other cutting conditions constant, increases the shear strain and specific energy for chip formation. Likewise, it has been observed that the shearing stress increases rapidly when the feed is reduced beyond a certain minimum, depending on cutting conditions.[14,15] Backer, Marshall, and Shaw have explained the phenomenon on the basis of the "size effect."

The authors propose another explanation which is interesting but in the writers' opinion requires some speculative assumptions, and the validity of the conclusions must be limited by the validity of the assumptions. The analogy introduced to compare the actual energy absorption by the workpiece in a cutting process with an assumed case illustrated in Fig. 13 of the paper, is based on certain assumptions which are very questionable. The mathematical calculation of the so-called equivalent force F_e' involves a further estimate of the surface displacement though it was acknowledged that reliable quantitative data for the determination of such plastic flow at the surface of the workpiece were not possible to obtain.

The authors state that when the depth of cut is varied from 0.0001 to 0.008 in. (which represents a change of 80 to 1), the surface displacement varies only from 0.003 to 0.005 in. (an increase in the ratio of 1.7 to 1). However, no explanation is given for the method which was used to obtain these values. A clear explanation of "surface displacement" and its measurement would be desirable.

During the past several years, considerable metal-cutting data have been collected in the Mechanical Engineering Department at the University of Illinois. The effect of feed on the cutting force and thrust force for conventional turning of steel under conditions where a type 2 chip is formed, in general, can be represented by Fig. 22 of this discussion. The results are for turning mill-annealed NE9445 steel with triple-carbide tools. Curves showing the same trend for spheroidized SAE 52100 steel have been reported earlier.[15] It is seen that the cutting force does vary approximately linearly with feed. However, this does not necessarily mean that extrapolation to extremely small feeds is permissible. Even at a cutting speed of 377 sfpm, which is considerably higher than that used by the authors, the chips produced showed evidence of significant built-up edge at feeds less than 0.0025 in. per revolution. The relationship between the thrust force and the feed departs appreciably from being linear. Although it might be argued that the cutting conditions in Fig. 22 are not strictly orthogonal, the discrepancy should be small in view of the high ratios of the depth of cut to the feed used.

Our experience in the machining of low and medium-carbon steels (both conventional and orthogonal cutting) with HSS tools at cutting speeds similar to those used by the authors (40–100 sfpm) is that a large built-up edge often exists, particularly in the range of small feeds. If this were the case, any expectation of correlation with Merchant's plasticity equation would be most optimistic.

As shown in Figs. 1 through 5 of the paper, and as stated by the

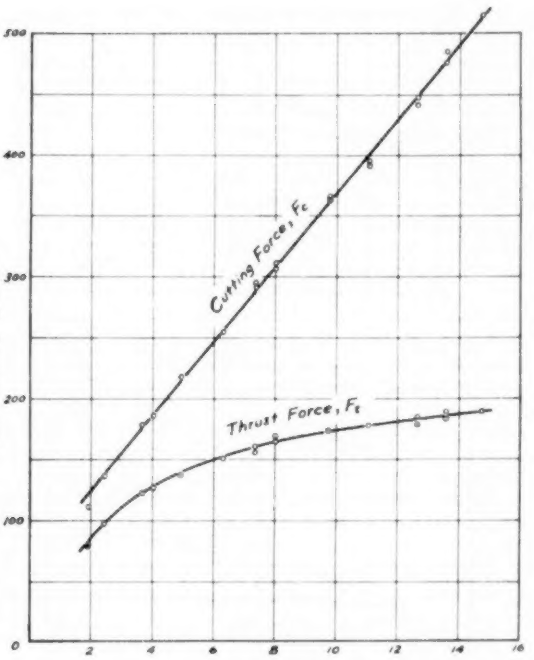
[11] Compare Fig. 5, Trans. ASME, vol. 73, 1951, p. 70, and figs. 3 and 4, "Residual Stresses in Machined Surfaces,"[8] p. 7.

[12] Professor of Mechanical Engineering, University of Illinois, Urbana, Ill. Mem. ASME.

[13] Assistant Professor of Mechanical Engineering, University of Illinois.

[14] Authors' Bibliography (5).

[15] "Thermophysical Aspects of Metal Cutting," by B. T. Chao, K. J. Trigger, and L. B. Zylstra, Trans. ASME, vol. 74, 1952, pp. 1039–1054.



Work material:	NE9445, mill-annealed
Tool material:	Triple carbide
Tool shape:	0-4-7-7-8-0-$\frac{3}{64}$
Cutting speed:	377 sfpm
Cutting depth	0.101 in.

Fig. 22 Effect of Feed on Tool Forces in Turning Steel

authors, the depth of workpiece deformation and magnitude of surface displacement appear to increase with greater depth of cut. Conclusion 2 of the paper based on extrapolation to zero depth of cut indicates that the force required to deform the workpiece is constant. The two are inconsistent.

The writers have submitted these comments as helpful suggestions in future studies. The authors have made a worth-while contribution in attempting to correlate the effect of deformation energy absorbed by the workpiece and to incorporate this heretofore ignored effect into metal-cutting theory. It is hoped that they will continue the investigation.

M. C. Shaw[16] and K. L. Chien.[17] The authors have presented an interesting discussion concerning the interpretation of force measurements in metal-cutting operations. In this discussion it is suggested that a significant part of the cutting force may be associated with the deformation of a layer of material on the finished surface. It is stated that this additional force may be responsible for a part of the increase in energy required to remove a given volume of material from the workpiece as the size of the layer removed is decreased (the size effect). A deeper insight into this problem may be had by considering it from a different point of view, as follows:

As the authors have indicated, a straight line with a positive intercept on the ordinate is always obtained when the cutting force is plotted against depth of cut in ordinary metal-cutting operations. Representative curves of this sort are shown in Fig. 23, herewith, for data previously published by Lapsley, Grassi, and Thomsen.[18] The source of this intercept may be ascribed to two causes:

1 To the size effect.

2 To the component of the cutting force associated with the deformation of the finished surface.

It has been found[14] experimentally that the energy required to cut a unit volume of metal u, increases significantly with a decrease in the depth of cut t, according to the following relation

$$u = \frac{K}{t} - u_0 \qquad [10]$$

where K and u_0 are constants. In any metal-cutting operation

$$u = \frac{F_H}{bt} \qquad [11]$$

where F_H is the cutting force and b is the width of cut. Combining Equations [10] and [11] we have

$$F_H = K_1 - K_2 t \qquad [12]$$

where K_1 ($= bK$) and K_2 ($= bu_0$) are constants. This equation is seen to be in agreement with the curves in Fig. 23 of this discussion, and hence if the rising energy relation that has been observed with decreased chip thickness is due to a size effect, then Equation [12] and the intercepts in Fig. 23 follow directly.

If, on the other hand, the intercept in Fig. 23 was interpreted as an extraneous force associated with plastic deformation in the finished surface, we might compute the shear and normal stresses on the shear plane using the net cutting force obtained by subtracting the intercept from the measured value of force in each case. This procedure is equivalent to saying that there is no size effect in the turning region and that all of the intercept is rather associated with plastic flow in the finished surface.

When the stress and strain obtained in the normal fashion[14] are

[16] Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass.

[17] Research Engineer, RCA Manufacturing Company, Camden, N. J.

[18] "Correlation of Plastic Deformation During Metal Cutting With Tensile Properties of the Work Metal," Trans. ASME, vol. 72, 1950, p. 979.

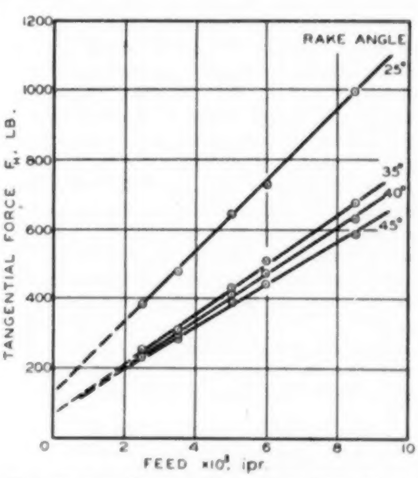


Fig. 23 Variation of Cutting Force With Feed Rate in Turning

compared with tensile test data, results similar to those in Fig. 24, herewith, are obtained. Here the cutting data[18] are represented by the marked points. The curve marked 1 is the ordinary true stress-strain curve for a specimen of the same material converted to shear co-ordinates by use of the maximum-shear theory (infinitesimal strain theory being employed). The second

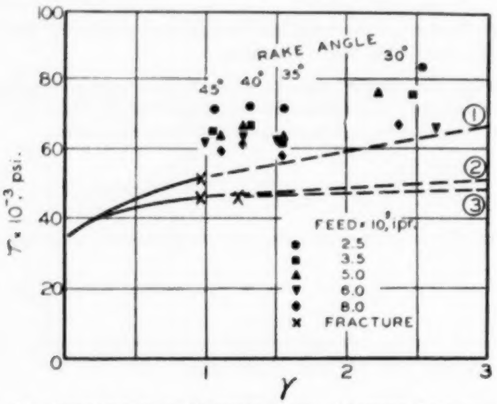


FIG. 24 COMPARISON OF CUTTING DATA—MARKED POINTS—AND TENSILE DATA

(Curve 1 corresponds to true stress-strain converted to shear co-ordinates using maximum-shear theory and infinitesimal-strain theory. Curve 2 is derived from curve 1 by making correction necessitated by neck which develops in tensile specimen. Curve 3 is derived from curve 2 by use of finite-strain theory.)

curve shows the result of correcting curve (1) for the triaxiality in the neck of the tensile specimen. The manner of making this correction has been suggested by Bridgman[19] and verified by Marshall and Shaw.[20] The third curve shows the effect of assuming the strains to be finite rather than infinitesimal. Curve (3) is the one which should describe the shear stress-shear strain characteristics of a material subjected to uniaxial shear as in the case of cutting. However, while all points are observed to lie above this line, the discrepancy that is noted is an orderly one—those points corresponding to a large depth of cut being lowest. This discrepancy has been attributed to the size effect. The foregoing procedure is equivalent to assuming all of the intercepts in Fig. 23 to be due to the size effect.

If we now recompute the data in Figs. 23 and 24, using the second technique, which assumes all of the intercepts in Fig. 23 to be associated with the additional cutting force involved in plastically deforming the finished work surface, we obtain the points shown plotted in Fig. 25. The discrepancy between curve (3) and the plotted points is now less than in Fig. 24. However, a size effect is still evident, although not as pronounced as in the first analysis. Thus we see that even when we assume there is no size effect, i.e., when none of the intercepts is assumed due to size effect, it is found that a size effect is still somewhat in evidence. This may be taken to mean that in reality at least part of the intercept shown in Fig. 23 is due to the size effect. It is not possible at this time to say just how much of the intercept is due to the size effect and how much, if any, is due to surface deformation.

Similar data are shown in Fig. 26, herewith. Here cutting and converted tensile data presented in another paper are shown. The numbers above each point refer to the depth of cut, while

[19] "Studies in Large Plastic Flow and Fracture," by P. Bridgman, McGraw-Hill Book Company, Inc., New York, N. Y., 1952.

[20] "The Determination of Flow Stress From a Tensile Specimen," by E. R. Marshall and M. C. Shaw, Trans. American Society for Metals, vol. 44, 1952, pp. 705–725.

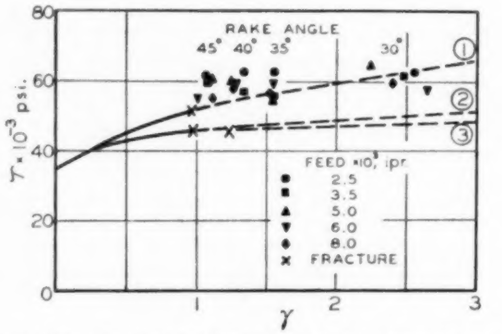


FIG. 25 RECALCULATION OF POINTS IN FIG. 24, ASSUMING ALL OF INTERCEPT IN FIG. 23 TO BE ASSOCIATED WITH ADDITIONAL FORCE TO CAUSE PLASTIC DEFORMATION IN FINISHED SURFACE

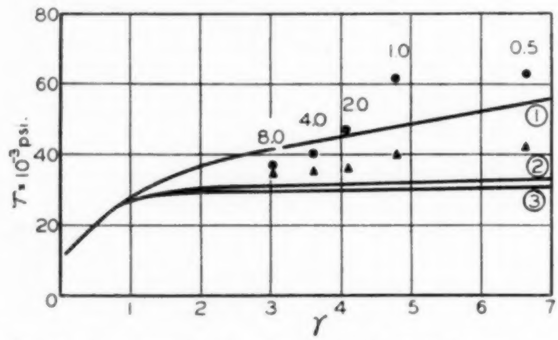


FIG. 26 COMPARISON OF CUTTING DATA—MARKED POINTS—AND TENSILE DATA

(Numbers above points refer to depths of cut in 0.001-in. units. Circles mark cutting data as measured, while triangles refer to data computed on basis that all of intercept on vertical axis, when F_H is plotted against depth of cut, is due to deformation of finished surface.)

the encircled numbers have the same significance as in Figs. 24 and 25. The circles mark the cutting data obtained by use of the observed cutting force in its entirety while the triangular data points refer to the corrected cutting forces assuming all of the intercept to be associated with surface deformation. The same relative situation is obtained in this case as before. A size effect is still evident with the triangular points and hence at least part of the intercept may be attributed to this cause.

Toward the end of their paper the authors state that the correlation between cutting and tensile data is poor inasmuch as some of the cutting data lie above the tensile-stress curve. This is not the case, however, when the Bridgman correction for triaxiality in the neck of the tensile specimen is made. The correction associated with the difference between the infinitesimal and finite strain theories is seen to be much less than the Bridgman correction. It is found that in cutting as well as in other problems associated with large plastic deformations, the maximum-shear theory gives better results than does the distortion-energy theory.

AUTHORS' CLOSURE

The authors are gratified by the comprehensive discussion the paper has elicited and they wish to thank the discussers for the interest they have displayed and the excellent analyses they have presented. An attempt will be made to answer questions raised by each discusser in the following.

The question of energy dissipation in the form of heat at the tool-chip interface raised by Dr. Hahn and its effect on F_c'' cannot

be answered precisely. It is believed that essentially all of the energy absorbed by the metal-cutting process is friction work, irrespective of whether it is plastic-flow or rubbing work, and hence appears in the form of heat. Varying the cutting speed has the effect of changing both the strength of these heat sources in the cutting zones, as well as the rate at which heat is being dissipated. Thus the temperatures of the metal in the chip, the tool, and the workpiece surface appear to be functions of speed and depend on the approach to isothermal or adiabatic conditions that may prevail during cutting. Thus F_c'', which contributes to both chip deformation and friction work at the tool-chip interface, must be a function of speed, but this functional relationship is not known.

The cutting data shown in Fig. 16 are interesting and show that under condition of extremely small depth of cut (chip too thin to be continuous) an entirely different relationship exists and hence the intercept correction becomes invalid. The authors also found a similar instability for chip thicknesses less than about 0.002 in. as shown by Dr. Hahn in Fig. 16. Their experimental values obtained for steel SAE 1113 are shown in Fig. 9. Of these data the cutting time for a depth of cut of 0.001 in. was short, since it was difficult to maintain the cutting action without tool chatter. This consideration, however, in no way proves that the cutting force becomes nonlinear with decreasing depth of cut for small chips, providing the chips were of the continuous type.

Professor Henriksen has given an interesting analysis in support of the authors' hypothesis that workpiece deformation may not be inappreciable during metal cutting. They agree that the process of deformation of a surface layer may be thought of as a friction process, in which the coefficient of friction usually is so high that under the presence of certain forces, as shown in Fig. 19, plastic deformation occurs. A similar plastic-deformation process is believed to occur along the tool face as the chip slides over the face and this mechanism is generally accepted as a friction process.

The authors also considered the possibility of calculating the forces causing plastic deformation or friction work from the residual stresses remaining in the surface of the workpiece. They discarded this idea, however, because they could not see that these residual elastic stresses could be used to calculate the deformation work, since this work is indeterminate unless the extent and kind of plastic deformation is known. This consideration does not exclude the possibility, however, that the resultant of the residual stresses may possibly be equal to the force which caused plastic deformation. If this were the case, the force can be calculated from residual stresses alone even though the deformation work cannot be obtained in a like manner.

If the force producing the surface deformation in the workpiece varies with depth of cut as that shown in Fig. 18, then the intercept correction clearly is only partial. Further investigation of this variation should be of interest and value.

It is unfortunate that an error appeared in the mimeographed preprints released at the time of presentation in Chicago. The lowest depth of cut for steel SAE 1113 was 0.001 in. instead of 0.0001 in. as reported in that preprint. The depth-of-cut ratio was thus 8 to 1 instead of 80 to 1 as assumed by Professors Trigger and Chao.

With respect to the comments by Professors Trigger and Chao concerning the speculative nature of the assumptions necessary to calculate the work for displacement of a surface layer of the workpiece, the authors believe the results are conservative since the deformation process was considered to be that of simple shear deformation. In reality, however, the process is more complex, as pointed out in the paper, and therefore would require higher forces than shown for equivalent surface displacement.

Professor Shaw and Mr. Chien have presented an interesting analysis in which they attempt to show that the intercept of the cutting force is probably made up of two effects, namely, size effect and workpiece deformation. The comparison they have made and have shown in Figs. 25 and 26 appears to leave no doubt that the size effect is still operative but probably to a smaller extent than heretofore believed. It is also of interest that the maximum finite shear theory appears to correlate better with metal-cutting data and tensile data than the distortion energy theory.

The authors are, in general, in agreement with the discussion that the intercept method is not an accurate method of correcting cutting forces and not all of this intercept force is due to deformation of the finished surface. They believe, however, that the approximate method of correction proposed by them will be useful in analyzing cutting data and will give a truer picture of the forces acting on the chip when corrections are made, especially for moderate depth of cut.

While extrapolation to extremely small depths of cut may not be permissible, as pointed out by these discussers, it in no way invalidates the use of the intercept correction method for the experimental data reported in this investigation providing it is taken as a first approximation.

Stress Relaxation in Compression of Rubber and Synthetic-Rubber Vulcanizates Immersed in Oil

By J. R. BEATTY[1] and A. E. JUVE,[1] BRECKSVILLE, OHIO

Stress relaxation in compression in air has been studied for compounds of various polymers. However, rubber seals, gaskets, and other applications are widely used where swelling agents, such as oil, are encountered. It was thought desirable to measure the stress-relaxation properties of various rubbers to determine their performance under such conditions. The results show that continuous stress relaxation is inhibited by the presence of oil and that unconfined swelling measurements predict the degree of inhibition directly. The variables of temperature and type of oil were investigated.

Introduction

Rubber and synthetic rubberlike materials, owing to their elastic properties, are particularly suited for use in gaskets, seals, suspensions, and other uses in essentially compressive strain. The stress resulting from this strain is widely utilized as a seal for fluids or gases. Many of these applications are subject to swelling agents, such as oil, from their surroundings either through design or accident. An earlier report (1)[2] described the stress-relaxation properties of seven rubber and synthetic-rubber compounds tested in air as a function of the variables of deformation, temperature, sample shape and size, and sample slippage. Swell phenomena have been investigated (2) with rubber compounds in the presence of swelling agents, but in these tests no strain was imposed on the rubber, and the effect of strain, particularly compression, has been reported only to decrease the rate of imbibition of oil (3).

To investigate the effect of mineral-oil swelling agents on rubber compounds four of the stocks were selected from the previous work (1). These were (*a*) a natural-rubber stock cured with TMTD; (*b*) a GR-S stock; (*c*) a neoprene GN stock; and (*d*) a Hycar OR-15 stock cured with TMTD. All were 60 ± 5 Shore "A" hardness, loaded with semireinforcing furnace black. The continuous stress relaxation of the samples was followed as a function of time, in the presence of three different mineral oils.

Testing Equipment

The tester used was that previously described (1) with only a slight modification. It consists essentially of a method of compressing a sample a predetermined percentage of its original height and measuring the minimum stress necessary to maintain this deflection as a function of time by the adjustable dead-weight loading. Fig. 1 is a diagram of the tester and the testing jigs

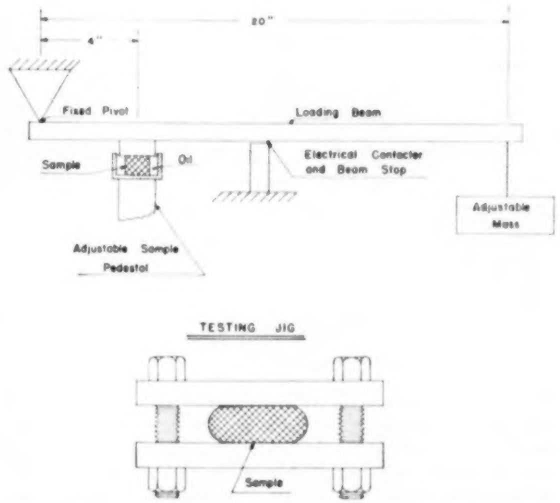


Fig. 1 Compression Stress-Relaxation Tester

used in long-time tests. The modification consisted of the substitution of an oil cup for the bottom platen of the adjustable sample pedestal which was used for short-time tests. The same jigs used in the preceding work were used exclusively for long-time tests during which they were stored immersed in the oil at the test temperature.

The test samples were ASTM compression-set samples 1.129 in. diam and 0.5 in. height having a loaded area of 1 sq in. The samples were die-cut from 6-in. × 10-in. sheets of the cured rubber. Recipes, cures, and stress-strain properties of the compounds tested in this work are given in Table 1. Table 2 lists properties of oils (4) used in this investigation covering the range of swelling tendency encountered in usual petroleum-base oils.

Test Procedure

Different procedures were adopted depending on whether the oil cup or the jigs were used. For the former, samples were conditioned in the oil cups immersed in oil until equilibrium temperature was attained, then deformed the desired amount and the stress measured as a function of time. The initial stress is obtained approximately from the load-deflection curve of a duplicate sample. Time is measured from imposition of load and the first value of stress recorded is at 36 sec or 0.01 hr which is referred to as the "zero time" in computing the relative stress values for S_t/S_0 where S_t is the stress at time t. Stress values were determined in approximately a geometric progression of time intervals since stress relaxation is approximately a logarithmic function of time.

When jigs were used the sample was conditioned in air with the sample dipped in oil before placing between the jig plates in order

[1] B. F. Goodrich Company Research Center.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Rubber and Plastics Division and presented at the Fall Meeting, Chicago, Ill., September 8–11, 1952, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, August 11, 1952.

TABLE 1 RECIPES, CURES, AND TENSILE PROPERTIES OF COMPOUNDS TESTED

Compound 1		Compound 2	
Natural rubber	100	GR-S10	100
Zinc oxide	5	Zinc oxide	5
SRF black	80	MBTS[b]	1.75
AgeRite powder	1	SRF black	65
TMTD[a]	3.5	Sulphur	1.75
	189.5		173.5
Cured 35 min at 140 C		Cured 90 min at 150 C	

Compound 3		Compound 4	
Neoprene GN	100	Hycar OR-15	100
SRF black	45	Zinc oxide	5
Zinc oxide	5	SRF black	50
Mag oxide	4	TMTD[a]	4
	154		159
Cured 40 min at 150 C		Cured 40 min at 150 C	

Physical Properties

Compound	Shore "A" durometer	Tensile strength, psi	Elongation, per cent	300 per cent modulus, psi
1	61	2800	425	1650
2	64	2400	460	1600
3	66	2700	420	2300
4	63	2600	520	1700

[a] Tetramethyl thiuram disulphide. [b] 2-2 Benzothiazole disulphide.

TABLE 2 TEST OILS

General description	Low swelling	Medium swelling	High swelling
Aniline point (ASTM Meth D611)	123.9 ± 1 C	93 ± 3 C	69 ± 1 C
Saybolt Universal Viscosity—Sec (ASTM Meth D 88)	98 ± 5	100 ± 5	155 ± 5
Flash point	279 C minimum	281.6 ± 5.5 C	201 ± 3 C

[a] ASTM Standards on Rubber Products.

that slippage between the rubber cylinder and the plates will take place immediately on loading. The test was then conducted in air for about the first 10 hr, after which it was removed from the tester with the initial compression maintained and immersed in the test oil at the desired temperature. The periodic checks of the stress were made in air, also for convenience. It was found that results obtained in this manner agreed with values obtained when the oil cup was used in which the sample was immersed continuously. Fig. 2 is a plot of data secured under both conditions of test.

The accuracy of the measurements using the oil-cup container is ±0.5 lb and as it is rare for the stress to decay below 100 lb the maximum error in the determinations is in the order of 1 per cent. With the jigs used in the long-time tests a possible source of error was introduced as the distance between the plates when the specimen is compressed is important. Extreme care was used in maintaining the distance constant, and duplicate tests in the oil cup, where the sample was continuously in the testing machine under constant deformation, show no major error results as long as a single operator makes all the measurements using the same technique.

Results of Tests

Effect of Type of Oil. In Figs. 3 to 10, inclusive, are the data showing the effect of the three oils at room temperature and 70 C for the compounds of natural rubber, GR-S, neoprene GN, and Hycar OR-15, the recipes of which are given in Table I.

Effect of Polymer. The data for the compounds of the four polymers tested in ASTM No. 3 oil are plotted in Fig. 11. Two examples of each class of the easily swollen (natural rubber and GR-S) and swell-resistant rubbers (neoprene GN and Hycar OR-15) are included.

Effect of Temperature. Tests were made at various temperatures. Table 3 gives the results for the four compounds in the low and high swelling oils, ASTM Nos. 1 and 3. Fig. 12 shows the accelerating effect of increasing temperatures. It is representative of the effect with other rubbers and oils.

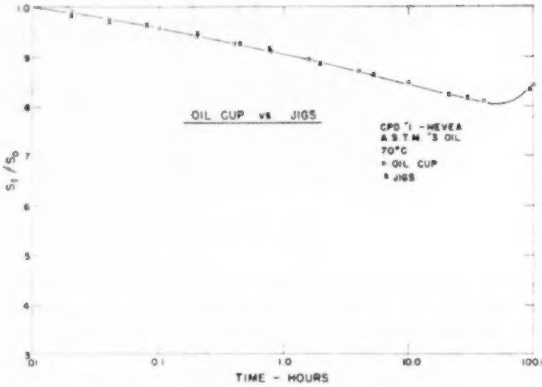


Fig. 2 Oil Cup Versus Jigs

Correlation of Volume Swell and Stress Increase. Volume swell was determined by ASTM method D471-15T (4) on the compounds of the four polymers in the three oils at 70 C for various times up to 1440 hr. Results are shown in Table 4 and Fig. 13. From the best-fitting smooth curve for each combination of compound and oil the 1440-hr volume-swell values were selected and are plotted in Fig. 14 as a function of per cent stress increase relative to the values in air at 1440 hr for the stress-relaxation tests.

Discussion of Results. In many applications the deterioration of physical properties of rubber brought about by contact with oil severely limits their application and/or service life as swollen rubbers are usually weak in tensile strength and low in modulus. The process by which this deterioration occurs is probably twofold. The solvent action weakens molecular attraction forces in the rubber while the double bonds are stretched which may lead to accelerated oxygen attack (2).

In intermittent stress-relaxation tests in which the sample is allowed to swell unconfined between measurements of stress at constant deformation, the stress relaxation is accelerated by the presence of oil (5). However, continuous stress-relaxation tests show a different behavior. In some cases, measurements of stress have shown that after 5000 hr test time the values exceeded the initial stress, and were always equal to or greater than the comparable measurements in air. In these tests the data are practically identical for an induction period which varies with the temperature and polymer, regardless of whether oil or air is in contact with the samples. After this induction period the action of the oil becomes apparent as an increase in stress above the values obtained in the presence of air. This increased stress was observed for times up to 10,000 hr.

Effect of Oil. The three oils used in this investigation cover the range of swelling normally encountered in usual petroleum-base oils. In Figs. 3 to 6 are shown the data for continuous stress relaxation with the samples immersed in the various oils at 24 C and 70 C of natural rubber and GR-S. Both increase considerably in all three oils and, for comparison purposes, results in air for the identical compounds from previous work are included. In every case ASTM No. 1 oil caused the smallest stress increase, ASTM No. 2 was intermediate, and ASTM No. 3 produced the greatest stress increase.

The oil-resistant-type rubbers, neoprene and Hycar OR-15, were unaffected by ASTM oils as shown in Figs. 7 and 9. They were practically unaffected by the ASTM No. 1 oil with respect to stress-relaxation properties at 70 C while ASTM Nos. 2 and 3 caused an appreciable increase in stress after considerable time as shown in Figs. 8 and 10.

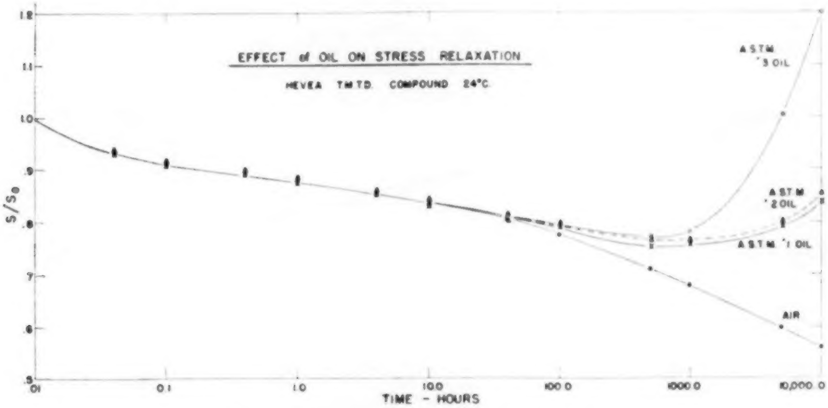


Fig. 3 Effect of Oil on Stress Relaxation
(Hevea TMTD compound 24 C.)

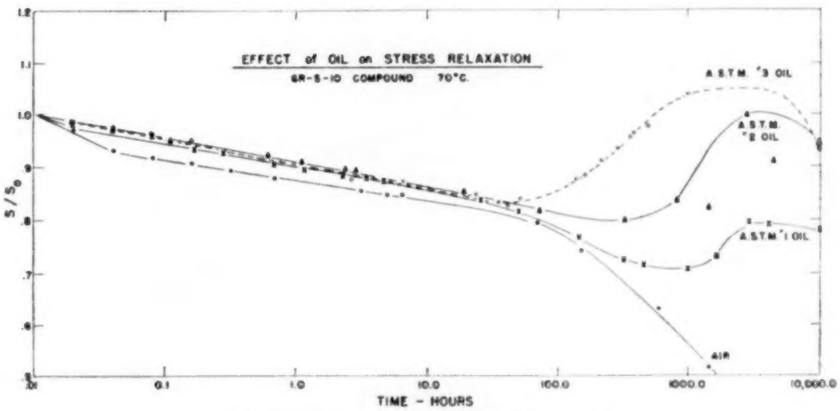


Fig. 4 Effect of Oil on Stress Relaxation
(GR-S-10 compound 70 C.)

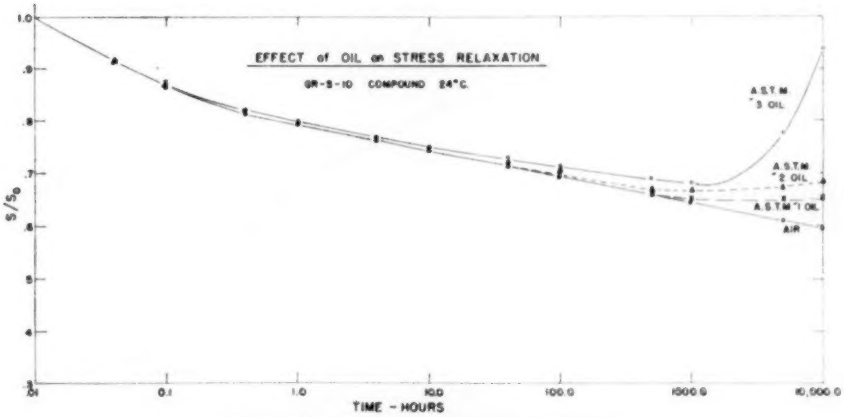


Fig. 5 Effect of Oil on Stress Relaxation
GR-S-10 compound 24 C.)

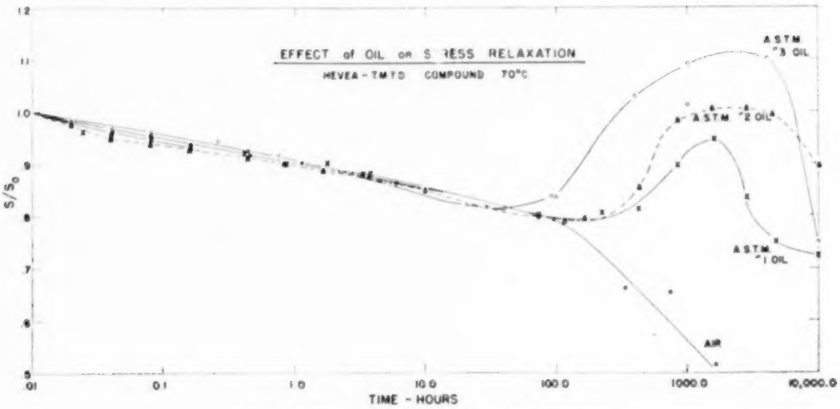


Fig. 6 Effect of Oil on Stress Relaxation
(Hevea-TMTD compound 70 C.)

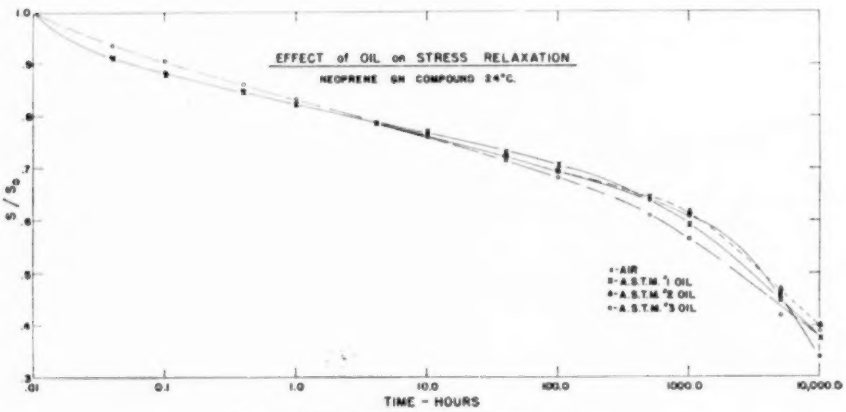


Fig. 7 Effect of Oil on Stress Relaxation
(Neoprene GN compound 24 C.)

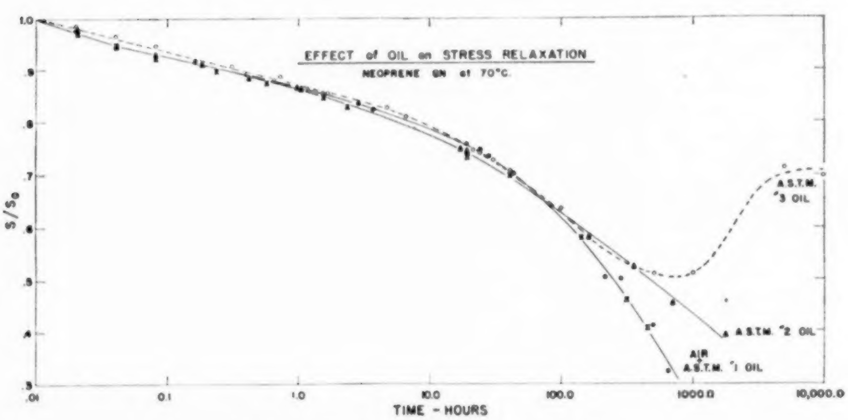


Fig. 8 Effect of Oil on Stress Relaxation
(Neoprene GN at 70 C.)

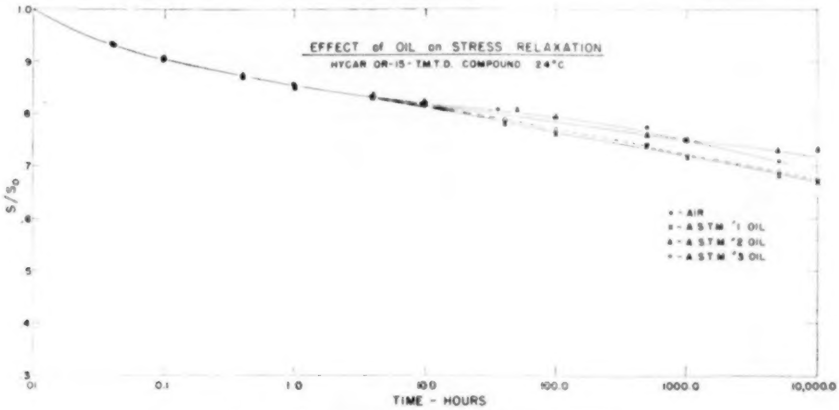


FIG. 9 EFFECT OF OIL ON STRESS RELAXATION
(Hycar OR-15-TMTD compound 24 C.)

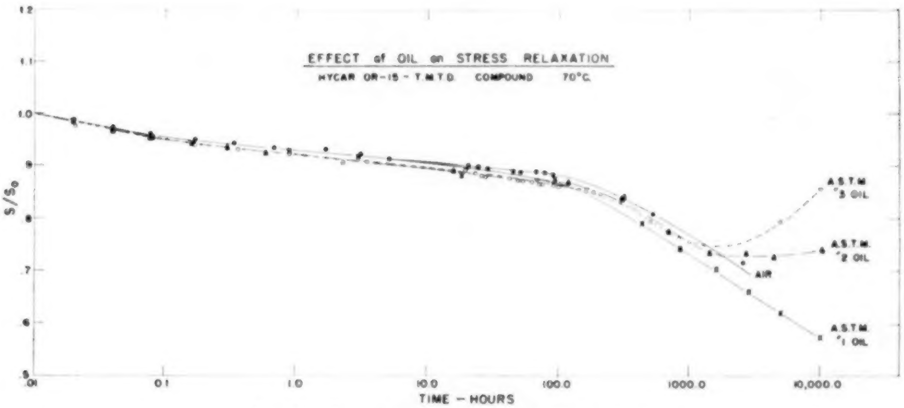


FIG. 10 EFFECT OF OIL ON STRESS RELAXATION
(Hycar OR-15-TMTD compound 70 C.)

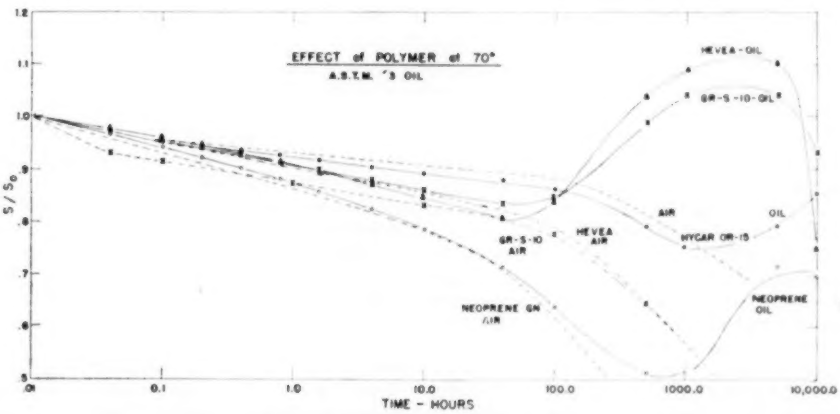


FIG. 11 EFFECT OF POLYMER AT 70 F
(ASTM No. 3 oil.)

TABLE 3 EFFECT OF TEMPERATURE

Time, hr	ASTM No. 1 oil — S/S_0 at temperatures of 24 C	70 C	90 C	ASTM No. 3 oil — S/S_0 at temperatures of 24 C	45 C	70 C	90 C
				Compound 1			
0.01	1.000	1.000	1.000	1.000		1.000	1.000
0.1	0.910	0.940	0.955	0.910		0.958	0.955
1	0.878	0.905	0.918	0.878		0.908	0.918
10	0.838	0.850	0.858	0.838		0.848	0.868
100	0.794	0.795	0.842	0.795		0.841	0.920
1000	0.756	0.927	0.900	0.780		1.090	1.022
5000	0.798	0.750		1.055		1.110	
10000	0.840	0.726		1.200		0.750	
				Compound 2			
0.01	1.000	1.000	1.000	1.000	1.000	1.000	1.000
0.1	0.867	0.940	0.925	0.870	0.910	0.952	0.925
1	0.792	0.911	0.875	0.800	0.868	0.905	0.875
10	0.744	0.855	0.790	0.750	0.823	0.860	0.790
100	0.700	0.785	0.700	0.715	0.780	0.850	0.752
1000	0.650	0.710	0.700	0.689	0.738	1.040	0.760
5000	0.647	0.790		0.775		1.040	
10000	0.650	0.780		0.940		0.930	
				Compound 3			
0.01	1.000	1.000		1.000	1.000	1.000	1.000
0.1	0.880	0.917		0.905	0.910	0.940	0.925
1	0.822	0.855		0.822	0.843	0.860	0.830
10	0.770	0.790		0.760	0.776	0.788	0.675
100	0.703	0.610		0.680	0.675	0.637	0.475
1000	0.590	0.270		0.565	0.450	0.512	0.500
5000	0.450	0.238		0.420		0.712	
10000	0.375			0.390		0.697	
				Compound 4			
0.01	1.000	1.000		1.000	1.000	1.000	
0.1	0.905	0.948		0.905	0.940	0.950	
1	0.850	0.909		0.850	0.900	0.912	
10	0.815	0.895		0.815	0.882	0.892	
100	0.715	0.870		0.770	0.850	0.862	
1000	0.720	0.735		0.725	0.800	0.755	
5000	0.682	0.620		0.690		0.795	
10000	0.672	0.575		0.675		0.855	

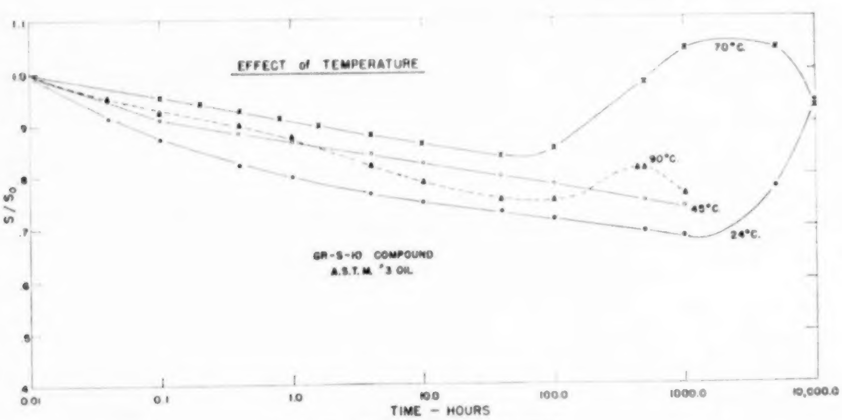


Fig. 12 Effect of Temperature (GR-S-10 compound ASTM No. 3 oil.)

The relatively long time for the onset of stress increase is in agreement with other investigators (6) who found that mechanical stress applied so as to resist the expansion of the rubber reduces the amount of oil absorbed and thus the swell.

Effect of Polymer. Polymers are affected by oil, depending on their molecular constitution. Polar groups or halogen groups contribute to the oil resistance in proportion to the amount present. Natural rubber and butadiene-styrene copolymers, such as GR-S, are not noted for their resistance to oils, as they are easily swelled in poor swelling agents. Comparative data for compounds of natural rubber, GR-S 10, neoprene GN, and Hycar OR-15 are shown in Fig. 11, for ASTM No. 3 oil at 70 C. The rubbers were rated in the following order: Hycar OR-15 showed the least change, neoprene considerably more, and GR-S and natural rubber the most. For comparison purposes comparable curves obtained in air for the same polymer, compound, and temperature conditions are shown as dotted lines. The magnitude of the action of the oil on the rubber is then apparent.

Effect of Temperature. Table 3 summarizes the results for the effect of temperature on stress relaxation, and Fig. 12 is representative for the rubbers in general. Increasing the temperature, in general, decreases the induction period and increases the rate and magnitude of stress increase above the values measured in air. In effect, the reactivity of the oil is increased with respect to the rubber and at the higher temperatures the oil-resistant rubbers are affected. This is due to increased solubility and decreased molecular attraction between molecules of the rubber.

TABLE 4 SWELL OF SR COMPOUNDS IN OIL

Time, hr	Volume increase at 70 C, per cent ASTM No. 1	ASTM No. 2	ASTM No. 3
Cpd. 1 (NR tuads cure)			
24	41	63	123
48	56	85	137
168	73	114	141
336	77	121	149
720	81	129	161
1440	92	139	181
Cpd. 2 (GR-S)			
24	19	35	93
48	23	50	113
168	35	81	121
336	37	98	121
720	39	91	125
1440	38	105	133
Cpd. 3 (Neoprene)			
24	4	12	35
48	5	17	49
168	9	31	70
336	10	39	75
720	31	45	81
1440	11	49	85
Cpd. 4 (Hycar CR-15)			
24	1.4	2	5
48	4.0	3	6
168	0.7	3	6
336	−7.0	7	7
720	0.9	2	7
1440	−2.0	7	6

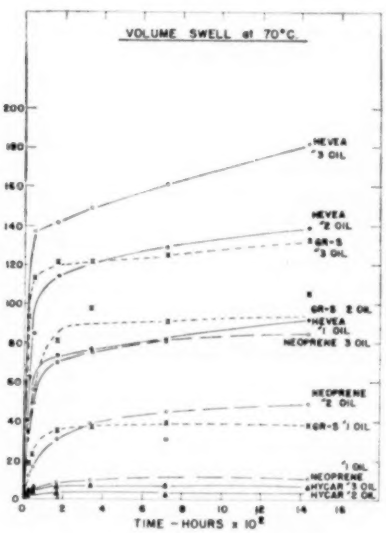


FIG. 13 VOLUME SWELL AT 70 C

Correlation of Volume Swell and Stress Relaxation. Neither volume swell nor increase in stress of the stress-relaxation tests has reached equilibrium at 1440 hr but the volume-swell curves appear to have reached a somewhat constant slope so it was assumed that a pseudo equilibrium had been attained in that the rates were probably of the same order of magnitude at this time. The significance of this correlation of stress increase and volume swell (shown in Fig. 14) is that the former may be predicted from the data of the latter, more easily performed, test.

The practical implications of this study are that rubber compounds which swell in the presence of oil have a property which may be utilized in some applications where it may serve a useful purpose. Examples are O-ring seals, and other types of gaskets where the rubber is used in compression. In these cases the

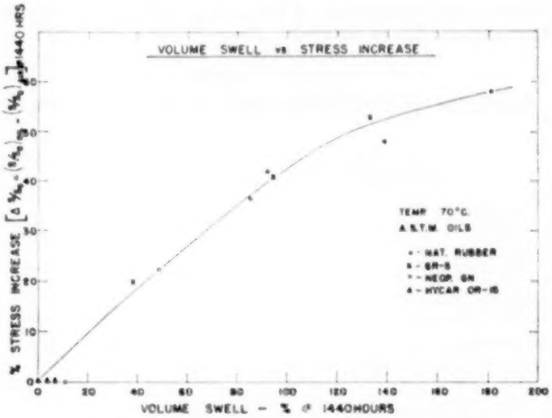


FIG. 14 VOLUME SWELL VERSUS STRESS INCREASE

stress decays more slowly with time and in some cases the force would increase and the tendency to leakage would be minimized. In these experiments the sample was relatively unconfined except for the direction of loading with only low frictional forces which tended to prevent increase in volume. It was noted that with natural rubber and GR-S at 70 C the stress reached a maximum between 1000 and 10,000 hr which is a result of the sample reaching equilibrium with respect to swelling by the oil and the stress then decreases depending on the oxidative scission of bonds in the same manner as found for tests conducted in air. However, according to Scott (2) the attack of swelling agents accelerates oxidation so it is possible that this oxidative scission might be in addition to that normally measured in air (3).

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Discussion

F. C. THORN.[3] The paper describes the effect of oil absorption in counteracting relaxation. Are data also available as to the effect of confinement in counteracting oil absorption? In other words, are there figures for the weight or volume increase in the

[3] Research Director, The Garlock Packing Company, Palmyra, N. Y. Mem ASME.

samples at the conclusion of the test, as compared to similar figures for unconfined samples?

Authors' Closure

Qualitative data indicated essentially no volume increase of the samples under compression until the stress began to decay rapidly between 5,000 and 10,000 hours for natural rubber and GR-S at 70 C. A volume increase, i.e., a diameter increase was noted at this interval. This may be compared with Fig. 13 which shows the volume change of unconfined rubber specimens.

Mechanical Properties of Glass-Cloth Plastic Laminates as Related to Direction of Stress and Construction of Laminate

By FRED WERREN,[1] MADISON, WIS.

Methods are available for predicting the directional properties of glass-cloth plastic laminates from a knowledge of the properties associated with their natural axes. The mechanical properties of cross-laminated or composite laminates may also be calculated from the basic properties of parallel laminates made of the component fabrics. Tests at the Forest Products Laboratory have shown that the suggested methods are reasonably correct for the range of glass-cloth plastic laminates presently being used in aircraft.

Introduction

In studies of the mechanical properties of wood, the Forest Products Laboratory has developed mathematical theory concerning these properties on the basis that wood is an orthotropic material, that is, a material having three mutually perpendicular planes of elastic symmetry. Under certain conditions, plywood also can be considered to be orthotropic, as are paper and paper laminates. Reinforced plastic laminates, made of sheets of woven fabric and resin, are also generally orthotropic, as are most of the core materials used in the construction of sandwich material for aircraft.

The development, verification, and application of the mathematical theory were accelerated during World War II, particularly as applied to plywood for use in aircraft. Much of this information was obtained in co-operation with the Army-Navy-Civil Committee on Aircraft Design Criteria and has subsequently been published as an ANC bulletin (1).[2]

Since the mathematical treatment for wood and plywood, which has been verified by tests on these materials, is general in nature, it should be applicable also to other orthotropic materials, such as glass-cloth laminates. The following discussion presents applicable equations and relates them to tests of glass-cloth plastic laminates for (*a*) the directional properties in tension, compression, and shear when no buckling occurs (2, 3), and (*b*) the mechanical properties of composite or cross-laminated, glass-cloth plastic laminates (4). Acceptable equations for calculating these properties are given.

Derivation of Equations

The mathematical theory of elasticity has been used extensively in the derivation of equations dealing with the behavior of orthotropic materials. When such a material is subjected to a single stress, the stress may be resolved into stresses associated with the natural axes. Thus the stress at proportional limit, for example, is assumed to occur when any one of the components of stress (resolved in the direction of the natural axes) reaches its proportional limit.

In the determination of strength, which is greater than the elastic limit of the material, an interaction formula of the second power has been suggested (5). The stresses are likewise resolved into stresses associated with the natural axes, and the strength of the material is assumed to be reached when these stresses satisfy the interaction equation. Thus, if the properties parallel to the natural axes are known, the properties at some angle to the natural axes may be determined.

More recently, a theory for the strength of orthotropic materials, based on the Henky-von Mises theory for isotropic materials, has been developed (6). The principal equation of this theory is similar to the interaction formula except for one additional term. This additional term has little effect on the theoretical values, particularly if the shear strength is low, as is the case for most reinforced laminates.

In a structure or in a test in which an orthotropic material is subjected to a single stress, two conditions should be considered. In one case, the material, or element, may be considered as not restrained, as in a long tensile specimen in which the center is free from end restraints. In another case, the element may be considered as completely restrained, a condition that may be approached, for example, by a wide compressive specimen when restraint is imposed by the heads of the testing machine. Equations are derived for both conditions. Under actual service or test conditions, the element may be subjected to some condition intermediate between these two. It will be seen, however, that the equations for the nonrestrained condition are more conservative than those for the restrained condition.

Several methods were tried for predicting the mechanical properties that are associated with the natural axes of cross-laminated and composite laminates from the properties of parallel laminates made of the component fabrics (4). The simplest method, a summation of the properties of the component plies, appeared to check the test values as well as any method tried.

Directional Properties

Consider an element of a laminate with its natural axes (A, B) as shown in Fig. 1. Stresses f_x, f_y, and f_{xy} may be applied at any angle θ to the natural axes. Based on the mathematical theory of elasticity, the following equations were derived (5):

Modulus of elasticity, E_x, and modulus of rigidity, μ_{xy}:

Tension or compression—no restraint

$$\frac{1}{E_x} = \frac{1}{E_a}\cos^4\theta + \frac{1}{E_b}\sin^4\theta + \left[\frac{1}{\mu_{ab}} - 2\frac{\sigma_{ab}}{E_a}\right]\sin^2\theta\cos^2\theta \qquad [1]$$

Tension or compression—complete restraint

$$\lambda E_x = E_a\cos^4\theta + E_b\sin^4\theta + (4\lambda\mu_{ab} + 2E_a\sigma_{ba})\sin^2\theta\cos^2\theta \qquad [2]$$

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[1] Engineer, Forest Products Laboratory, Forest Service, U. S. Department of Agriculture. The laboratory is maintained in co-operation with the University of Wisconsin.

[2] Numbers in parentheses refer to Bibliography at end of paper.

Contributed by the Rubber and Plastics Division and presented at the Fall Meeting, Chicago, Ill., September 8–11, 1952, of The American Society of Mechanical Engineers.

 Manuscript received at ASME Headquarters, July 10, 1952. Paper No. 52—F-36.

Shear—no restraint

$$\frac{1}{\mu_{xy}} = \frac{1}{\mu_{ab}}\cos^2 2\theta + \left[\frac{1}{E_a} + \frac{1}{E_b} + 2\frac{\sigma_{ab}}{E_a}\right]\sin^2 2\theta \ldots [3]$$

Shear—complete restraint

$$\mu_{xy} = \mu_{ab}\cos^2 2\theta + \frac{1}{4\lambda}(E_a + E_b - 2E_a\sigma_{ba})\sin^2 2\theta \ldots [4]$$

in which

μ_{ab} = modulus of rigidity of laminate associated with shearing strain in A-B plane

σ_{ab} = Poisson's ratio of contraction of laminate parallel to O-B, Fig. 1, to extension parallel to O-A associated with tension parallel to O-A

σ_{ba} = Poisson's ratio of contraction of laminate parallel to O-A, Fig. 1, to extension parallel to O-B associated with tension parallel to O-B

$\lambda = 1 - \sigma_{ab}\sigma_{ba}$, and is near unity for most laminates

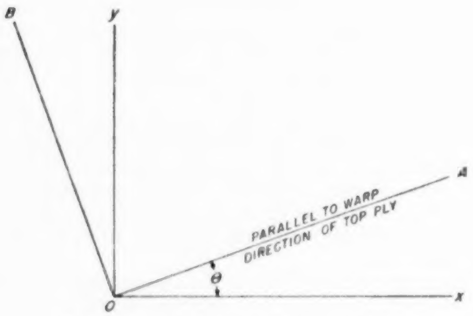

FIG. 1 ILLUSTRATION OF CHOICE OF AXES

When computing tension values, E_a and E_b are the tensile moduli associated with the natural axes; that is, $\theta = 0°$ and $\theta = 90°$. The corresponding compressive moduli are used in computing the compression values.

In shear, the meanings of symbols E_a and E_b change as the value of θ changes from negative to positive. When θ is negative, E_a and E_b are the moduli of elasticity in compression parallel and tension perpendicular to the warp direction of the face ply. When θ is positive, E_a and E_b are the moduli of elasticity in tension parallel and compression perpendicular to the warp direction. If $\theta = -45°$, for example, the warp direction of the face ply is subjected to compressive stress, while the fill direction is subjected to tensile stress.

***Proportional Limit,* p_x *and* p_{xy}.** Since it is assumed that the proportional limit will be reached when any one of the components in tension, compression, or shear reaches its proportional limit, the applicable component is the one that gives the least value:

Tension or compression—no restraint

$$\left.\begin{aligned} p_x &= \frac{p_a}{\cos^2\theta} \\ p_x &= \frac{p_b}{\sin^2\theta} \\ p_x &= \frac{-p_{ab}}{\sin\theta\cos\theta} \end{aligned}\right\} \ldots\ldots [5]$$

Tension or compression—complete restraint

$$\left.\begin{aligned} p_x &= \frac{p_a\lambda E_x}{E_a(\cos^2\theta + \sigma_{ba}\sin^2\theta)} \\ p_x &= \frac{p_b\lambda E_x}{E_b(\sin^2\theta + \sigma_{ab}\cos^2\theta)} \\ p_x &= \frac{-p_{ab}E_x}{2\mu_{ab}\sin\theta\cos\theta} \end{aligned}\right\} \ldots\ldots [6]$$

Shear—no restraint

$$\left.\begin{aligned} p_{xy} &= \frac{p_a}{2\sin\theta\cos\theta} \\ p_{xy} &= \frac{-p_b}{2\sin\theta\cos\theta} \\ p_{xy} &= \frac{p_{ab}}{\cos^2\theta - \sin^2\theta} \end{aligned}\right\} \ldots\ldots [7]$$

Shear—complete restraint

$$\left.\begin{aligned} p_{xy} &= \frac{p_a\lambda\mu_{xy}}{E_a(1-\sigma_{ba})\sin\theta\cos\theta} \\ p_{xy} &= \frac{-p_b\lambda\mu_{xy}}{E_b(1-\sigma_{ab})\sin\theta\cos\theta} \\ p_{xy} &= \frac{p_{ab}\mu_{xy}}{\mu_{ab}(\cos^2\theta - \sin^2\theta)} \end{aligned}\right\} \ldots\ldots [8]$$

Terms p_a and p_b are taken from tension tests when computing tensile values and from compression tests when computing compression values. In shear, whether the tensile or compressive value is used depends upon the angle θ, as already shown. Because it is difficult to establish a proportional limit in shear at $\theta = 0°$, p_{ab} in these equations is taken as one half of the proportional limit in tension when $\theta = 45°$. This is permissible because this relationship is true, within the elastic range at that angle. The proper sign must be applied to the value of p_{ab}, since p_{ab} is numerically equal to $-p_{ab}$.

Maximum Stress, F_x *and* F_{xy}. During World War II, the Forest Products Laboratory suggested the use of an interaction formula of the second power to determine strength values at angles to the orthotropic axes (5). The condition for failure was given by the formula

$$\left(\frac{t_{aa}}{F_a}\right)^2 + \left(\frac{t_{bb}}{F_b}\right)^2 + \left(\frac{t_{ab}}{F_{ab}}\right)^2 = 1 \ldots\ldots [9]$$

in which

t_{aa} = direct stress in warp direction of top ply

t_{bb} = direct stress at right angles to warp direction of top ply

t_{ab} = shear stress parallel and perpendicular to warp direction of top ply

F_a = strength under direct stress acting in warp direction of top ply

F_b = strength under direct stress acting perpendicular to warp direction of top ply

F_{ab} = strength under shear stress acting parallel and perpendicular to warp direction of top ply

From this equation were derived other equations for determining strength values in tension, compression, and shear.

A theory for strength of orthotropic materials, based on the Henky-von Mises theory, was developed recently (6). The condition for failure is expressed by

$$\left(\frac{t_{aa}}{F_a}\right)^2 - \left(\frac{t_{aa}t_{bb}}{F_aF_b}\right) + \left(\frac{t_{bb}}{F}\right)^2 + \left(\frac{t_{ab}}{F_{ab}}\right)^2 = 1 \ldots\ldots [10]$$

An examination of Equations [9] and [10] shows that they are very nearly alike, particularly if the shear strength of the material is low. They differ only by the presence of the term $-\dfrac{t_{as}t_{bb}}{F_aF_b}$ in Equation [10].

Experience with the application of Equation [9] has shown that it is possible to predict reasonable values of strength at various angles of loading. The more exact theory may be applied if desired, but in some instances it results in equations that are quite cumbersome compared with those based on the interaction formula. The following equations, therefore, are those derived from the interaction formula.

Maximum Strength. Tension or compression—no restraint

$$\frac{1}{F_x^2} = \frac{\cos^4\theta}{F_a^2} + \frac{\sin^4\theta}{F_b^2} + \frac{\sin^2\theta\cos^2\theta}{F_{ab}^2} \qquad [11]$$

Tension or compression—complete restraint

$$F_x^2 = F_a^2\cos^4\theta + F_b^2\sin^4\theta + 4F_{ab}^2\sin^2\theta\cos^2\theta \qquad [12]$$

Shear—no restraint

$$\frac{1}{F_{xy}^2} = \frac{1}{F_{ab}^2}\cos^2 2\theta + \left[\frac{1}{F_a^2} + \frac{1}{F_b^2}\right]\sin^2 2\theta \qquad [13]$$

Shear—complete restraint

$$F_{xy}^2 = F_{ab}^2\cos^2 2\theta + \frac{1}{4}(F_a^2 + F_b^2)\sin^2 2\theta \qquad [14]$$

F_a and F_b are tensile or compressive strengths at $\theta = 0°$ and $\theta = 90°$. The selection of the proper value follows that outlined previously for other properties.

In the derivations of the strength equations for the condition of no restraint, it was assumed that the applied stress is the only stress acting. In the derivations of the equations for the condition of complete restraint, however, it was assumed that the two stresses other than the applied stress were not zero but had values that maximized the value the applied stress assumes at the condition of failure.

Correlation of Theoretical and Test Values. Laminates reinforced with one of three glass fabrics having entirely different properties were tested with respect to their directional properties. The fabrics included (*a*) a satin-weave fabric (181-114), (*b*) a plain-weave fabric (112-114), and (*c*) a unidirectional fabric (143-114). All three laminates were parallel-laminated with a polyester resin, and the correlations of test results with theoretical values were similar. The 143-114 laminate, with the greatest

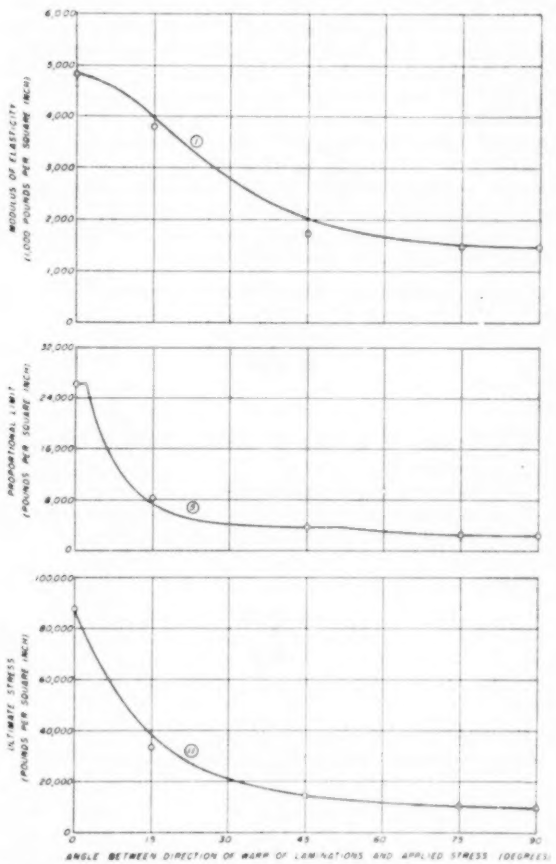

Fig. 2 Theoretical and Experimental Values From Tension Tests of 143-114 Laminate

(Equation number is given for each curve.)

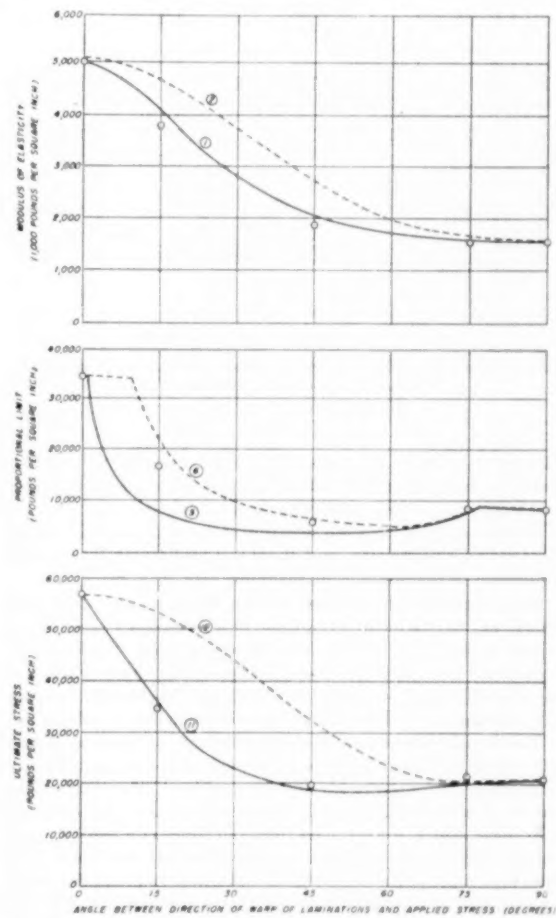

Fig. 3 Theoretical and Experimental Values From Compression Tests of 143-114 Laminate

(Equation number is given for each curve.)

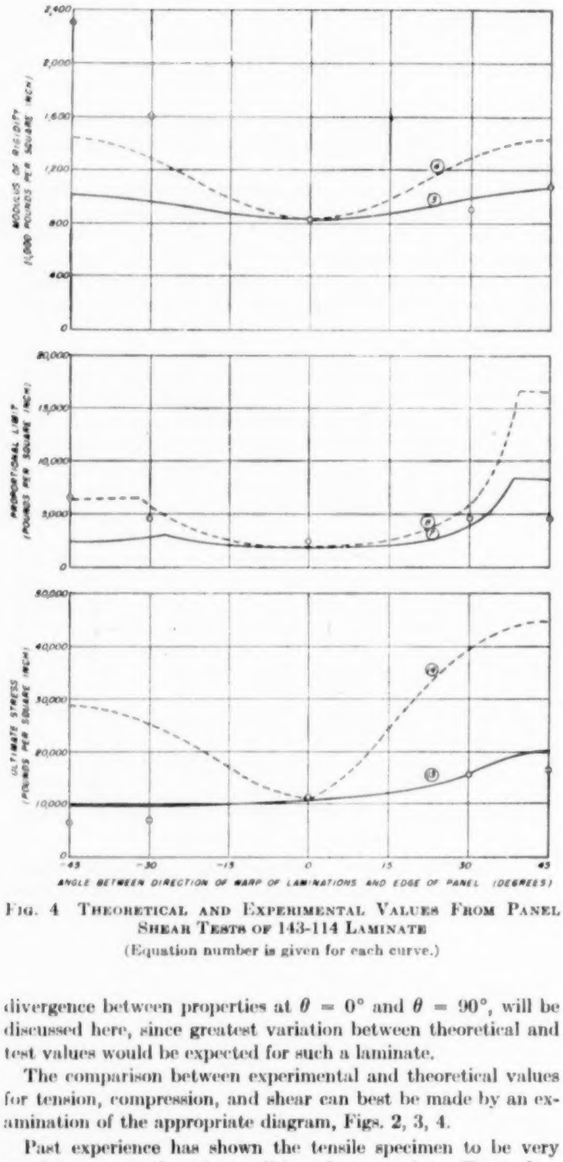

FIG. 4 THEORETICAL AND EXPERIMENTAL VALUES FROM PANEL SHEAR TESTS OF 143-114 LAMINATE

(Equation number is given for each curve.)

divergence between properties at $\theta = 0°$ and $\theta = 90°$, will be discussed here, since greatest variation between theoretical and test values would be expected for such a laminate.

The comparison between experimental and theoretical values for tension, compression, and shear can best be made by an examination of the appropriate diagram, Figs. 2, 3, 4.

Past experience has shown the tensile specimen to be very nearly representative of a condition of no restraint. Test values would be expected to fall near the appropriate "no-restraint" curve, and thus only the curves corresponding to a condition of no restraint are shown, Fig. 2. Very good correlation between the experimental values and the theoretical curves is evident.

Results of tests of compression specimens may vary considerably, depending upon the shape of the specimen and the restraint imposed on the ends by the testing machine. The compression values plotted in Fig. 3 are based on a specimen 1 in. wide and 4 in. long, which is restrained from buckling by a special apparatus. Strain measurements were made across the center 2 in. Just how much influence exists as a result of end restraint is not known, but experience has shown that specimens tested in this way approach the no-restraint condition. Fig. 3 shows good correlation between experimental values and the theoretical curves.

The difficulty of making an accurate shear test has long been recognized. Results of shear tests do not check the theoretical curves nearly as well as do those from the tension and compression tests, except at $\theta = 0°$. This lack of correlation is probably due to inadequacies in the method of test.

It has been recognized that the maximum stress in shear at 0°, F_{ab}, can be obtained for orthotropic materials from tension tests at 0°, 45°, and 90°. Thus values of F_{ab} may be calculated from Equation [11] by substituting values of F_0, F_{45}, and F_{90}. A similar relationship can be established for modulus of rigidity and proportional-limit stress through the use of tension values at 0°, 45°, and 90° properly substituted in Equations [1] and [5]. It is believed that the shear values obtained by this method are probably more nearly correct than those obtained from the use of the present type of panel-shear test. Such results, if based on tensile specimens that are long and not restrained by the grips, should provide a practicable method of obtaining the shear properties.

In the design of laminated structures or structural elements, the degree of restraint is usually indeterminate. It appears, then, that the more conservative no-restraint condition should be assumed.

CROSS-LAMINATED AND COMPOSITE LAMINATES

The properties of glass-cloth laminates may be varied by varying the orientation of the laminations or by combining laminations of different properties. The laboratory has tested cross laminates of three fabrics having varying ratios of strength in the warp and fill directions. Also tested was a parallel laminate combining two of these fabrics in alternate laminations. From these tests methods have been developed for predicting the properties associated with the natural axes, based on the properties of parallel laminates made of the component fabrics. By such a method it is possible to estimate the mechanical properties of any cross laminate or composite laminate from a knowledge of the properties of parallel laminates of the component materials.

The mechanical properties associated with the natural axes may be estimated by the following methods:

Modulus of Elasticity. Tension and compression

$$E = \frac{1}{A}\sum_{i=1}^{i=n} E_i A_i \qquad [15]$$

where

E = modulus of elasticity of cross-laminated or composite laminates
A = cross-sectional area of laminate
E_i = modulus of elasticity of ith ply in direction of stress
A_i = cross-sectional area of ith ply

Shear

$$G = \frac{1}{A}\sum_{i=1}^{i=n} G_i A_i \qquad [16]$$

where

G = modulus of rigidity of laminate
G_i = modulus of rigidity of ith ply in direction of stress

Static Bending

$$E = \frac{1}{I}\sum_{i=1}^{i=n} E_i I_i \qquad [17]$$

where

E = modulus of elasticity of laminate
I = moment of inertia of laminate about its neutral plane
E_i = modulus of elasticity of ith ply parallel to span
I_i = moment of inertia of ith ply about the neutral plane of the laminate

Strength Properties. Tension, compression, and shear

$$S = \frac{1}{A} \sum_{i=1}^{i=n} S_i A_i \qquad [18]$$

where

S = property of laminate
S_i = property of ith ply in direction of stress

Static Bending

$$S = \frac{1}{I} \sum_{i=1}^{i=n} S_i I_i \qquad [19]$$

Three cross-laminated panels and one composite panel were tested in tension, compression, shear, and static bending by the methods described in the Appendix. Comparable theoretical values were calculated by use of the equations given, using the mechanical-property values obtained from previous tests of parallel laminates. The ratios of the computed properties to the test properties are given in Table 1.

An examination of these ratios shows that the computed values of modulus of elasticity and of maximum strength are in reasonable agreement with the test values. Modulus-of-rigidity ratios vary considerably from unity, but it is probable that this variation results from inadequacies in the method of test and from the small number (two or three) of shear tests made on each material. The agreement of proportional-limit values is quite erratic in some instances, but this must be expected because it is usually quite difficult to establish the proportional limit exactly.

The methods of calculating the mechanical properties of cross-laminated and composite laminates given are applicable only at 0° and 90° to the warp direction. If properties at other angles are desired, they may be estimated by the application of the elastic and interaction formulas, since the materials may be considered as orthotropic.

TABLE 1 RATIO OF COMPUTED MECHANICAL PROPERTIES TO PROPERTIES OBTAINED FROM TESTS OF FOUR LAMINATES WITH STRESSES APPLIED AT 0° AND 90° TO WARP OF LAMINATIONS

Property	Laminate construction[a]							
	1		2		3		4	
	0°	90°	0°	90°	0°	90°	0°	90°
Tension								
Modulus of elasticity	1.09	1.03	1.02	1.02	1.08	1.12	0.95	0.92
Stress at proportional limit	1.19	1.13	1.07	1.06	1.16	1.28	0.90	0.91
Maximum stress	1.04	1.03	1.03	1.05	0.98	0.98	0.98	0.95
Compression								
Modulus of elasticity	0.96	0.94	0.94	0.98	1.02	0.97	0.99	0.95
Stress at proportional limit	0.96	1.23	1.23	0.92	1.23	1.10	0.86	0.98
Stress at 0.2 per cent offset	0.90	0.94	0.96	0.95	1.09	0.89	0.96	0.98
Maximum stress	0.90	0.94	0.96	0.96	1.09	0.90	0.96	1.01
Shear								
Modulus of rigidity	1.16	...	0.92	...	1.21	...	1.10	...
Stress at proportional limit	0.80	...	1.02	...	0.98	...	0.99	...
Stress at 0.2 per cent offset	0.90	...	0.96	...	0.94	...	0.94	...
Maximum stress	0.94	...	1.01	...	0.86	...	1.07	...
Static Bending								
Modulus of elasticity	1.00	0.99	0.97	0.97	1.06	0.98	0.96	0.98
Stress at proportional limit	0.96	1.05	0.88	0.99	1.37	1.31	0.91	1.37
Stress at 0.2 per cent offset	1.04	1.04	0.95	1.00	1.01	0.86	0.97	1.22
Modulus of rupture	1.04	1.04	0.96	1.00	1.02	0.91	0.97	0.95

[a] Construction 1, 112-114 fabric, cross-laminated.
Construction 2, 116-114 fabric, cross-laminated.
Construction 3, 143-114 fabric, cross-laminated.
Construction 4, Alternate plies of 112-114 and 143-114 fabrics, parallel-laminated.

All of the laminates were $^1/_8$ in. thick and made with a typical polyester resin

Conclusions

Equations based on the mathematical theory of elasticity and on an interaction formula can be used for the determination of the directional properties of orthotropic laminates. Such laminates include those reinforced with one or more glass fabrics, either parallel or cross laminated. Reasonable values of mechanical properties associated with the natural axes of cross-laminated or composite laminates may be predicted by the use of other suggested equations, which have been verified by tests of four glass-cloth plastic laminates of different constructions.

In view of the difficulty in making an accurate shear test, it is suggested that the shear properties associated with the natural axes be arrived at on the basis of the tensile properties at 0°, 45°, and 90°. Proper substitution of these values in the appropriate equation will result in values that are probably more nearly correct than those obtained by the panel-shear test.

Appendix

Methods of Test

Tensile specimens were 16 in. long and of the thickness of the laminate. The maximum sections at the ends were $1^1/_2$ in. wide

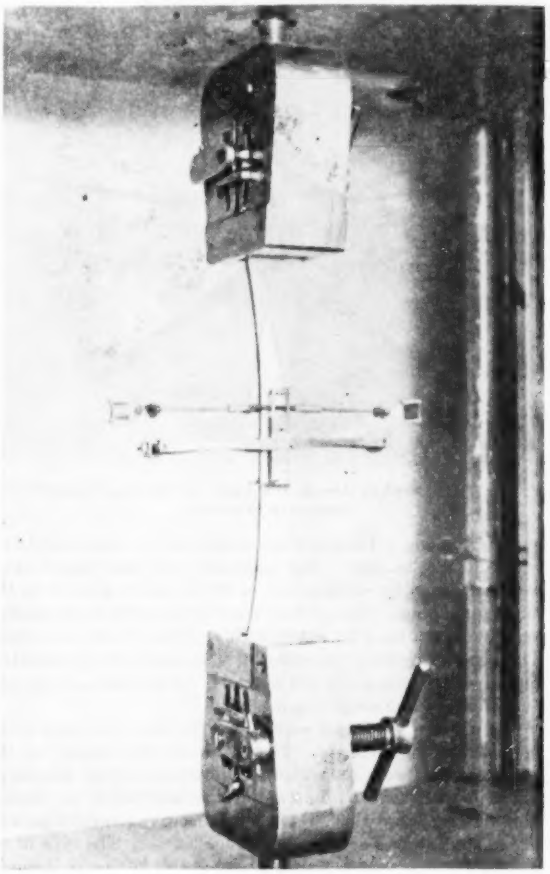

Fig. 5 Tensile Test Used in Testing Glass-Cloth-Laminate Specimens

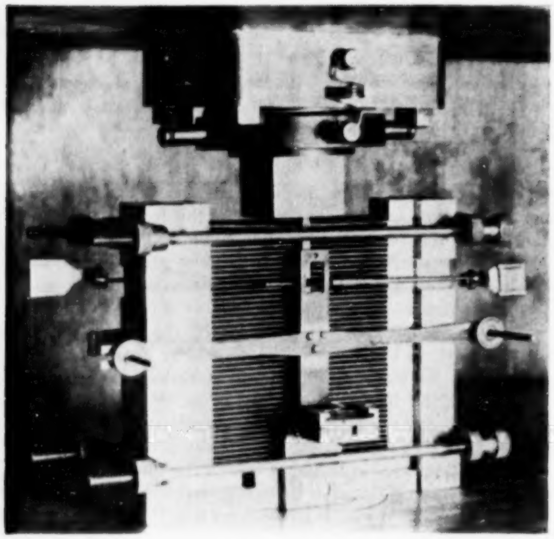

Fig. 6 Compression Pack Test of Type Used in Testing Glass-Cloth-Laminate Specimens

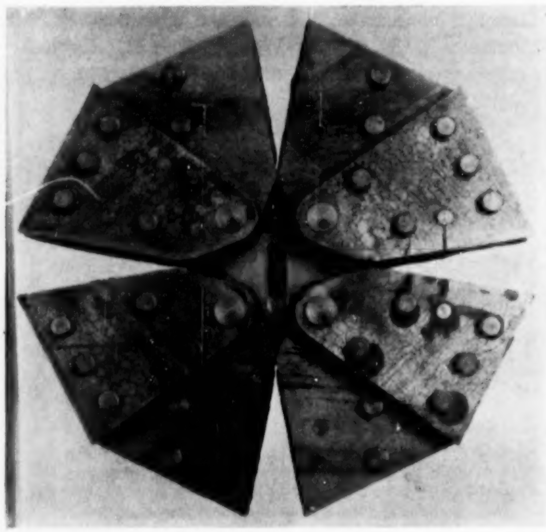

Fig. 7 Panel-Shear Apparatus Used in Testing Glass-Cloth-Laminate Specimens

and $2^7/_8$ in. long. The minimum section at the center was 0.8 in. wide and $2^1/_2$ in. long. The maximum and minimum sections were connected by circular arcs of 20 in. radius tangent to the minimum section. Strains were measured parallel to the applied load across a 2-in. gage length with a pair of Marten's mirrors. This type of specimen was selected to minimize stress concentrations and the influence of end restraint. A specimen, set up and ready for test, is shown in Fig. 5.

Compression specimens were 1 in. wide, 4 in. long, and of the thickness of the laminate. The specimens were loaded on the 1-in. ends and were restrained from buckling by an apparatus similar to that shown in Fig. 2 of the American Society for Testing Materials Standard D805-47 (7). Strains were measured parallel to the applied load across a 2 in. gage length. The type of restraining device used in testing plywood and laminates is shown in Fig. 6.

Panel-shear specimens were cut to the shape of a formee cross,

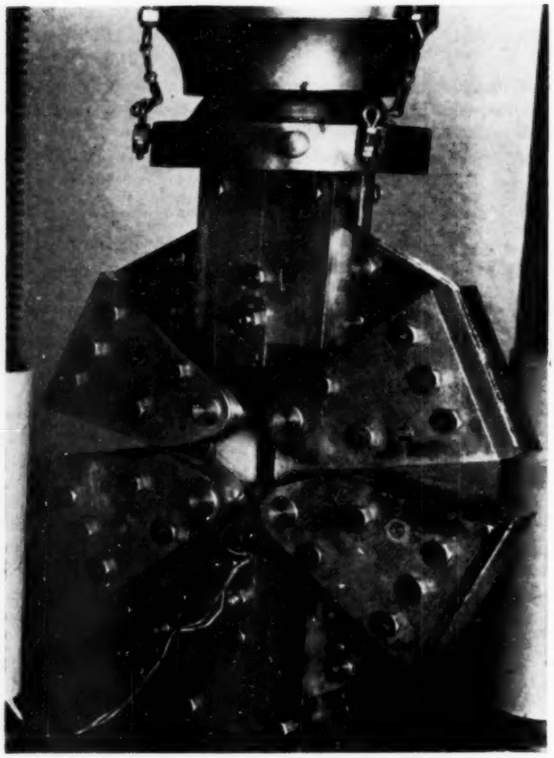

Fig. 8 Method of Testing Panel-Shear Specimen of Glass-Cloth Laminate

(Strain measurements can be made with metalectric gages as shown, or with mechanical gages employing dials.)

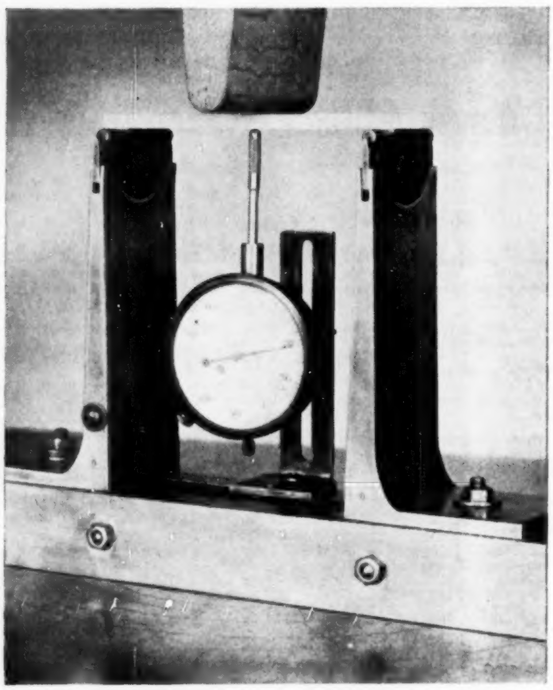

Fig. 9 Static-Bending Apparatus Used in Testing Glass-Cloth Laminates

the outline of which is shown in Fig. 7. The part of the specimen common to the four arms of the cross was 3 in. square. Four pairs of machined steel plates having the shape of the arms, excluding the 3-in. square at the center, were cemented to the arms. Upon completion of the bonding, machine bolts were added and tightened, and thus the specimen was clamped between steel plates. The load was applied in such a way as to direct the load along the edges of the 3-in-sq central section of the specimen. Thus an initial condition of approximately pure shear stress was obtained in this section. The apparatus is shown, set up ready for test, in Fig. 8. Compressive-strain measurements were made on each face at the center with a dial extensometer or metalectric strain gage.

Bending specimens were $^1/_2$ in. wide, 6 in. long, and of the thickness of the laminate. They were tested under center loading over a 4-in. span, and deflections were measured with a dial gage. A specimen, set up and ready for test, is shown in Fig. 9.

All laminates discussed in this paper were $^1/_4$ in. thick. The reinforcing fabric was combined with a laminating resin of the polyester (styrene-alkyd) type. The laminates are considered to be typical of laminates made with any polyester resin conforming with U. S. Air Force Specification 12049 and a specific glass fabric with finish 114.

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Effect of Preloading and Fatigue on Mechanical Properties of Glass-Cloth Plastic Laminates

By A. D. FREAS,[1] MADISON, WIS.

Tensile tests of glass-cloth plastic laminates showed that modulus of elasticity and proportional-limit stress may be changed considerably after the first application of stress. A small number of preloads, in either tension or compression, have no appreciable effect on the strength of the laminate. The application of many repeated or reversed stresses, however, may result in a fatigue strength that is much lower than the static strength of the material.

Introduction

Most glass-cloth plastic laminates, when loaded in tension, produce a load-deformation curve that exhibits two straight lines, which differ somewhat in slope. A question arises, therefore, as to the significance of the initial and secondary values of modulus of elasticity and stress at proportional limit and as to the effect on these values of loading to a stress below the secondary proportional limit. An aircraft structural part in service frequently will be loaded to a stress beyond the initial stress at proportional limit, and any effect of such preloading on the reported properties of a laminate would be of practical significance to the aircraft designer.

Preloading once or a few times to some stress level may be expected to affect such properties as modulus of elasticity and stress at proportional limit without necessarily having a significant effect on the ultimate strength. Frequently repeated or reversed loadings above certain minimum stress levels, however, are known to reduce the breaking strength below that which is measured in a static test.

Fatigue tests may be likened to preload tests, except that the preload tests reported herein are limited to a few preloads (generally only one), while the fatigue-test "preloads" are numbered in the thousands or millions. During their service lives, most aircraft structures are subjected to many repetitions or reversals of load. Some knowledge of the load level that repeatedly may be sustained without failure is therefore necessary to the intelligent design of aircraft structures.

This report presents the results of fatigue and preload tests to serve as a guide to the fatigue behavior of glass-fabric laminates. The tests were carried out by the Forest Products Laboratory in co-operation with the ANC-17 Panel on Plastics for Aircraft of the Subcommittee on Air Force-Navy-Civil Aircraft Design Criteria, Aircraft Committee, Munitions Board.

The laminates tested conformed to the requirements of U. S. Air Force Specifications 12049 and 12051. In addition, the fabrics used were of sufficient variety to cover the range likely to be used in aircraft applications. The results of the tests, therefore, may be considered typical of results to be expected from glass-fabric polyester laminates conforming with Air Force specifications.

Effect of Preloading[2]

Materials Tested. After normal conditioning of specimens, an initial series of tests was made on laminates fabricated from two representative fabrics. One, 181-114, was a satin-weave fabric having much the same properties in directions parallel and perpendicular to the warp direction. The other, 143-114, was a unidirectional fabric having markedly different properties in the two directions. Each was parallel-laminated with a laminating resin of the polyester (alkyd-styrene) type.

A small, supplementary series of wet and dry tests involved a parallel-laminated, fine-weave fabric (112-114); the same unidirectional fabric (143-114) as in the first series, cross-laminated instead of parallel-laminated; and a parallel-laminated, coarse-weave fabric (184-114). Parallel-laminated 181-114 fabric was repeated in this series to provide data after wet-conditioning.

Some significant characteristics of the various laminates are shown in Table 1.

TABLE 1 GENERAL PROPERTIES OF LAMINATES TESTED IN PRELOADING

Panel number	Fabric	Direction of plies	Total number of plies	Specific gravity	Resin content, per cent	Barcol hardness
First Series—Laminates $^1/_8$ In. Thick						
54	181-114	Parallel	12	1.81	34.2	68
70	181-114	Parallel	12	1.76	38.7	65
49	143-114	Parallel	13	1.85	31.3	66
69	143-114	Parallel	13	1.86	31.4	65
Second Series—Laminates $^1/_4$ In. Thick						
24	181-114	Parallel	23	1.79	36.0	70
25	143-114	Cross	26	1.87	31.0	70
9	112-114	Parallel	84	1.69	43.9	69
56	184-114	Parallel	9	1.87	29.9	67

Methods of Test. *Tension:* The tension specimens were 16 in. long and of the thickness of the laminate, with a reduced cross section at the center. The maximum sections at the end were $1^1/_2$ in. wide and $2^7/_8$ in. long. The minimum section at the center was 0.8 in. wide and $2^1/_2$ in. long and was connected to the end section by circular arcs of 20-in. radius.

The control specimens were loaded at a constant head speed of 0.035 ipm to failure. Deformations parallel to the applied load were measured across a 2-in. gage length by means of a pair of Marten's mirrors reading to 0.00001 in., except for the dry tests in the second series, in which a pair of 1-in. metalectric strain gages were used to measure strain. The preloaded specimens were loaded at a constant head speed of 0.035 ipm to a predetermined load below the estimated maximum. Load-deformation readings

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, July 10, 1952. Paper No. 52—F-35.

[2] "Effect of Prestressing in Tension or Compression on the Mechanical Properties of Two Glass-Fabric-Base Plastic Laminates," by Fred Werren, Forest Products Laboratory Report No. 1811, September, 1950, and Supplement A, June, 1951.

were taken at regular increments up to maximum preload. The load was then dropped off, and a few load-deformation readings were taken while unloading. The deformation reading at initial load was recorded as an indication of the amount of set to be expected. Most specimens were preloaded once and then run to failure. A few specimens were preloaded, the load released, and the specimens preloaded one or two additional times. In the final run, the specimen was loaded continuously to failure.

Compression: The compression specimens were 1 in. wide × 4 in. long and of the thickness of the laminate. They were loaded on the 1 in. ends at a head speed of 0.012 ipm and were restrained from buckling by being clamped lightly between spring-steel fingers.

The control specimens were loaded continuously to failure, deformations parallel to the applied load being measured over a 2-in. gage length by means of a pair of Marten's mirrors mounted on opposite edges of the specimens. Preloaded specimens were loaded by a procedure similar to that described for the tension specimens.

Conditioning: Specimens for dry tests were conditioned at 75 F and 50 per cent relative humidity. Specimens for wet tests were conditioned at 100 F and nearly 100 per cent relative humidity for 60 days prior to test.

Presentation and Discussion of Data. Load-deformation curves: Fig. 1 shows typical tensile load-deformation curves for specimens of 181-114 laminate, one specimen being preloaded to a stress between the initial and secondary proportional limits, and the second being loaded to a stress intermediate between the secondary proportional limit and the ultimate stress. It is evident that the short initial line formed during preloading is no longer present after preloading. Rather, if the maximum preload is between the initial and secondary proportional limits, the load-deformation curve consists of a straight line up to the maximum preload and a second straight line having the same slope as the secondary line of the first run. When the maximum preload is at or above the secondary proportional limit, the subsequent load-deformation curve is a straight line up to the secondary proportional-limit stress and breaks off from that point on.

Laminates made of other fabrics exhibited similar load-deformation curves. Additional applications of a preload after the first preload do not appear to change the load-deformation curve. This is evidenced by the left-hand group of curves in Fig. 2.

For laminates, such as parallel-laminated 143-114 fabric, that do not exhibit the dual straight lines when stressed in tension and for compression loadings where dual straight lines are not found, preloading does not appear to affect the form of the load-deformation curve. This is illustrated by the right-hand group of curves in Fig. 2 and by Fig. 3.

Modulus of elasticity: The tensile modulus of elasticity after preloading is intermediate between the initial and secondary moduli found from the first loading, Tables 2 and 3. The ratio of the final modulus to the secondary modulus varies with the magnitude of the preload, the final modulus approaching the secondary modulus with an increase in the level of preloading. In the tests reported herein, the final modulus ranged from about 3 to 14 per cent greater than the secondary modulus. It would appear conservative for purposes of design, therefore, to use the secondary modulus ordinarily reported.

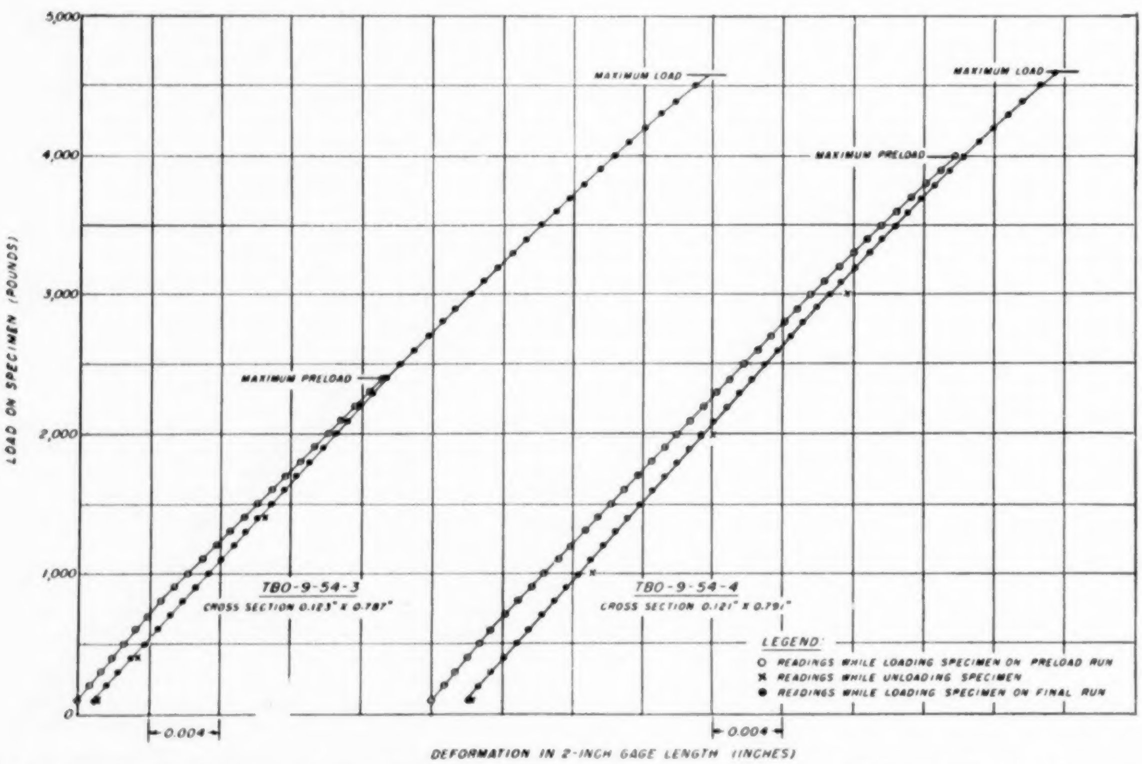

Fig. 1 Typical Load-Deformation Curves for Two Tensile Specimens, Nos. TBO-9-54-3 and TBO-9-54-4, of 181-114 Laminate
(Each specimen was loaded to maximum preload, load was removed to initial load, and then specimen was loaded to failure.)

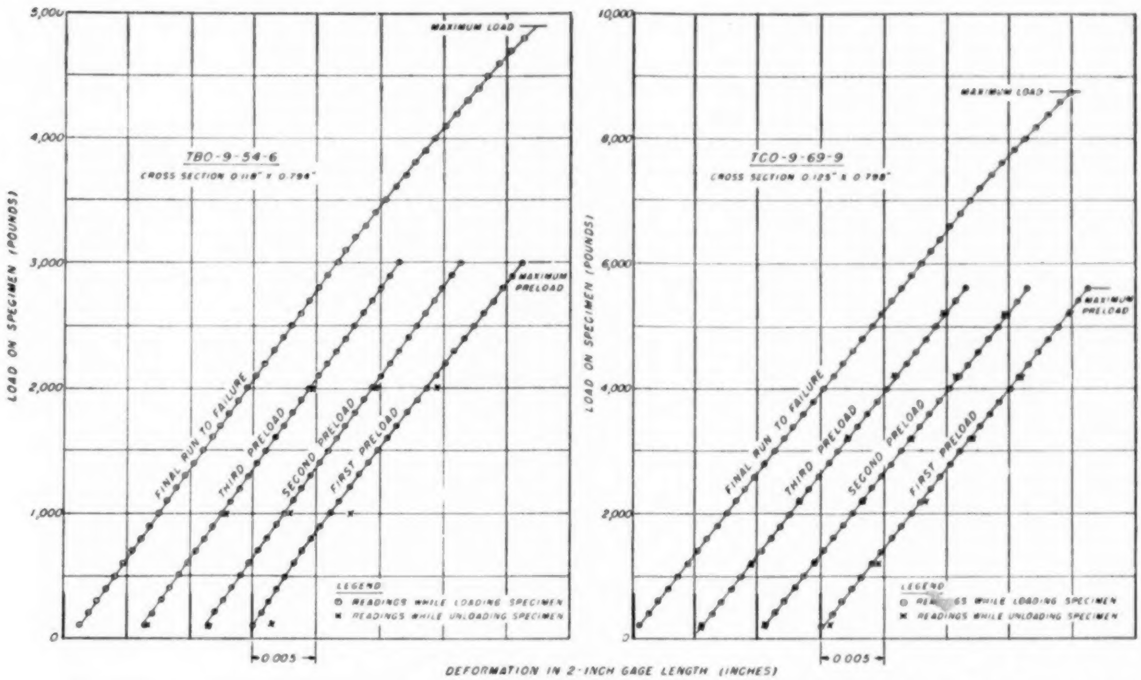

FIG. 2 (*Above*) TYPICAL LOAD-DEFORMATION CURVES FOR TENSILE SPECIMENS OF 181-114 LAMINATE, NO. TBO-9-54-6 (*Left*) AND 143-114 LAMINATE NO. TCO-9-69-9 (*Right*) SHOWING EFFECT OF PRELOADING SEVERAL TIMES ON LOAD-DEFORMATION CURVES

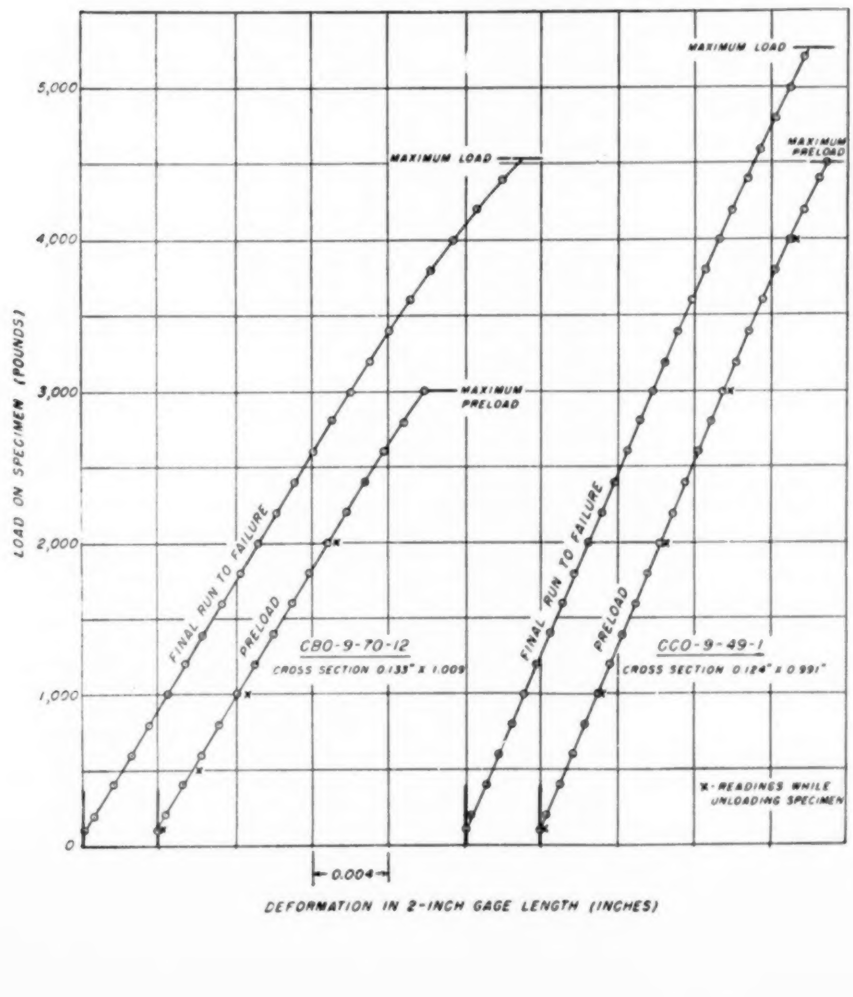

FIG. 3 (*Left*) TYPICAL LOAD-DEFORMATION CURVES FOR COMPRESSION SPECIMENS OF 181-114 LAMINATE, NO. CBO-9-70-12 (*Left*) AND 143-114 LAMINATE, NO. CCO-9-49-1 (*Right*)

(Each specimen was loaded once to maximum preload, load was removed to initial load, and then specimen was loaded to failure.)

TABLE 2 RESULTS OF TENSION TESTS OF 181-114 LAMINATE AND 143-114 LAMINATE, SHOWING EFFECT OF PRELOADING ON THE MECHANICAL PROPERTIES AND OBSERVED SET

Specimen No.	Thickness	Width	Number of preload runs	First run: Modulus of elasticity, Initial[1]	First run: Modulus of elasticity, Secondary[2]	First run: Proportional-limit stress, Initial[1]	First run: Proportional-limit stress, Secondary[2]	First run: 0.2 percent offset stress[3]	First run: Maximum applied stress	First run: Observed set in 2-inch gage length	Final run: Modulus of elasticity	Final run: Proportional-limit stress	Final run: Ultimate stress	Ratio of maximum applied stress to ultimate stress	Ratio of E of final run to secondary E of first run
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)
	Inch	Inch		1,000 p.s.i.	1,000 p.s.i.	P.s.i.	P.s.i.	P.s.i.	P.s.i.	Inch	1,000 p.s.i.	P.s.i.	P.s.i.	Percent	
181-114 Laminate — Control Specimens															
T80-9-54-2	0.123	0.795	....	3,020	2,760	7,160	36,820	46,790	46,790	....	....	....	46,790	100.0	....
54-5	.118	.795	....	3,320	2,870	6,400	35,180	47,440	48,610	....	....	....	48,610	100.0	....
54-8	.118	.791	....	3,020	2,770	7,920	34,270	51,080	51,100	....	....	....	51,100	100.0	....
70-2	.133	.798	....	3,710	2,560	6,970	30,150	39,570	46,640	....	....	....	46,640	100.0	....
70-5	.133	.799	....	2,980	2,650	5,650	28,230	42,820	46,580	....	....	....	46,580	100.0	....
70-8	.135	.799	....	3,020	2,700	6,500	32,450	44,960	46,350	....	....	....	46,350	100.0	....
Average..	....	....	....	3,180	2,720	6,770	32,850	45,440	47,680	....	....	....	47,680	100.0	....
181-114 Laminate — Preloaded Specimens															
T80-9-54-7	.117	.793	3	3,180	2,720	6,470	....	....	10,780	0.00067	3,100	11,860	50,550	21.4	1.14
54-1	.124	.796	1	2,970	2,670	7,090	....	....	18,240	.00130	2,970	18,240	48,330	37.8	1.11
70-6	.134	.795	2	3,250	2,430	7,510	....	....	19,710	.00141	2,920	19,710	46,140	42.7	1.20
70-4	.141	.797	1	2,960	2,450	5,340	....	....	19,580	.00100	2,690	19,580	44,400	44.1	1.10
54-3	.123	.787	1	3,100	2,640	8,780	....	....	24,790	.00110	2,920	24,790	47,470	52.3	1.11
70-3	.132	.792	1	2,800	2,630	6,700	....	....	25,830	.00099	2,790	25,830	46,580	55.5	1.06
54-6	.118	.794	3	3,420	2,800	7,470	....	....	32,020	.00148	3,070	32,020	52,250	61.4	1.10
70-9	.136	.800	3	2,920	2,610	6,890	24,820	....	35,850	.00171	2,870	20,220	46,880	76.5	1.10
54-9	.116	.799	1	3,430	2,990	7,010	30,210	....	38,840	.00169	3,210	30,210	50,710	76.6	1.07
70-7	.142	.801	1	2,960	2,470	8,350	25,500	....	35,170	.00165	2,640	22,860	44,220	79.5	1.07
54-4	.121	.791	1	3,130	2,760	6,270	31,340	....	41,790	.00213	2,940	31,340	48,170	86.8	1.07
70-1	.130	.789	1	2,870	2,470	6,820	30,220	38,510	41,920	.00196	2,600	30,220	46,020	91.1	1.05
Average..	....	....	....	3,080	2,640	7,060	28,420	....	....	....	....	....	47,640	....	....
143-114 Laminate -- Control Specimens															
T00-9-49-2	.123	.790	....	5,100	....	53,510	....	86,650	86,650	....	....	....	86,650	100.0	....
49-5	.124	.795	....	5,110	....	62,890	....	85,620	85,620	....	....	....	85,620	100.0	....
49-8	.125	.793	....	5,290	....	78,690	....	86,760	86,760	....	....	....	86,760	100.0	....
69-2	.120	.796	....	5,760	....	77,470	....	82,080	82,080	....	....	....	82,080	100.0	....
69-5	.122	.797	....	5,090	....	74,050	....	92,770	92,770	....	....	....	92,770	100.0	....
69-8	.125	.798	....	5,040	....	....	....	82,000	82,000	....	....	....	82,000	100.0	....
Average..	....	....	....	5,230	....	69,320	....	85,980	85,980	....	....	....	85,980	100.0	....
143-114 Laminate -- Preloaded Specimens															
T00-9-49-7	.125	.788	3	4,940	....	....	....	....	18,270	0.00026	4,940	52,790	89,340	20.5	1.00
69-4	.122	.795	1	5,370	....	....	....	....	24,740	.00007	5,370	59,800	85,160	29.0	1.00
49-3	.123	.793	3	5,300	....	....	....	....	32,810	.00026	5,300	65,610	81,400	40.3	1.00
69-3	.122	.795	1	5,460	....	....	....	....	39,180	.00019	5,460	63,920	87,230	45.0	1.00
49-1	.119	.780	3	5,660	....	....	....	....	45,250	.00037	5,660	62,490	84,030	53.9	1.00
49-6	.125	.793	1	5,380	....	....	....	....	52,460	.00025	5,380	60,530	84,340	62.2	1.00
69-9	.125	.798	3	5,160	....	....	....	....	56,140	.00063	5,160	72,180	87,020	64.5	1.00
69-7	.125	.797	1	4,770	....	54,200	....	....	60,230	.00037	4,770	54,200	87,330	69.0	1.00
49-9	.125	.788	1	5,110	....	....	....	....	64,970	.00091	5,110	71,070	85,690	75.9	1.00
49-4	.123	.794	1	5,280	....	....	....	....	73,720	.00056	5,280	79,870	87,850	84.0	1.00
69-6	.124	.795	1	5,280	....	....	....	....	68,980	.00051	5,280	....	78,620	87.7	1.00
69-1	.120	.795	1	5,260	....	62,890	....	....	79,660	.00053	5,260	62,890	86,060	92.5	1.00
Average..	....	....	....	5,250	....	....	....	....	....	....	....	....	85,340	....	....

[1] Values based on initial straight portion of stress-strain curve.

[2] Values based on second straight portion of stress-strain curve.

[3] Value at 0.2 percent offset from initial straight line of stress-strain curve.

Stress at proportional limit: As shown in Tables 2 and 3, the stress at proportional limit after preloading will be some value between the stresses at the initial and secondary proportional limits ordinarily reported, depending upon the degree of preloading. The proportional-limit stress will correspond with the preload up to a maximum equal to the secondary proportional-limit stress. For a higher preload, the proportional-limit stress will correspond with the secondary proportional-limit stress ordinarily reported.

There appears to be little if any difference between proportional-limit stresses in compression before and after preloading, Table 4.

Maximum stress: Maximum stress in tension and compression does not appear to be affected by preloading.

Hysteresis and set: The slight hysteresis indicated in Fig. 1 was typical of all laminates tested.

The observed set at initial load in the tests increased with an increase in the level of preloading. This is indicated by the curves in Figs. 4 and 5. It will be noted, however, that the observed set is relatively small, even for large preloads. It was found that repeated loading did not increase the amount of set appreciably.

Fig. 6 presents data illustrating the relation between set and offset (deviation of the load-deformation curve at the level of preloading from the initial straight-line portion of the curve) for the 181-114 laminate loaded in tension. The scatter of the data is sufficient to preclude the drawing of an average curve. It appears, however, that, with the exception of the three lowest points, a straight line from the origin would be reasonably representative of the relationship. Roughly, it appears that the observed set in tension is about one half as great as the offset for the 181-114 laminate.

Effect of conditioning: The behavior of a few tensile specimens tested after conditioning at high humidity is similar to that of the specimens tested in the dry condition, Table 3. The limited data indicate, however, that the difference between the modulus of elasticity after preloading and the secondary modulus of elasticity is smaller for specimens tested wet than for specimens tested dry. For example, one of the laminates, tested dry, showed a difference between these values of about 7 per cent

TABLE 3 RESULTS OF TENSION TESTS OF FOUR LAMINATES SHOWING THE EFFECT OF PRELOADING ON THE MECHANICAL PROPERTIES AND OBSERVED SET

Specimen No.	Thickness	Width	First run: Modulus of elasticity: Initial[1]	First run: Modulus of elasticity: Secondary[2]	First run: Proportional limit stress: Initial	First run: Proportional limit stress: Secondary	First run: Maximum applied stress	First run: Observed set in 2-inch gage length[3]	Final run: Modulus of elasticity	Final run: Proportional limit stress	Final run: Ultimate stress	Ratio of maximum applied stress to ultimate stress	Ratio of E of final run to secondary E of first run
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
	Inch	Inch	1,000 p.s.i.	1,000 p.s.i.	P.s.i.	P.s.i.	P.s.i.	Inch	1,000 p.s.i.	P.s.i.	P.s.i.	Percent	
DRY TESTS													
Laminate 143-114, Resin 2 -- Cross-laminated													
TCO-9-25-1	0.233	0.797	3,500	2,820	5,380		21,540	0.00051	3,040	19,390	54,280	39.7	1.08
2	.233	.794	3,400	2,770	6,490	32,430	37,840	.00126	2,920	32,430	53,840	70.4	1.05
Average			3,450	2,800	5,940	32,430					54,060		
Laminate 112-114, Resin 2 -- Parallel-laminated													
TAO-9-9-1	.250	.796	3,060	2,450	7,040		25,130	.00100	2,730	24,120	42,210	59.5	1.11
2	.253	.793	2,930	2,300	8,970	24,920	32,900	.00150	2,530	22,930	42,600	77.2	1.10
Average			3,000	2,380	8,000	24,920					42,400		
Laminate 184-114, Resin 2 -- Parallel-laminated													
TNO-9-56-1	.234	.794	3,500	3,000	7,530	25,830	30,140	.00083	3,130	27,990	52,640	57.3	1.04
2	.234	.793	3,460	2,970	7,540	28,020	37,720	.00124	3,090	28,020	53,350	70.7	1.04
Average			3,480	2,980	7,540	26,920					53,000		
WET TESTS													
Laminate 181-114, Resin 2 -- Parallel-laminated													
TBO-9-24-1	.241	.799	3,040	2,680	6,230		12,460	.00026	2,880	12,460	43,210	28.8	1.07
2	.241	.785	3,080	2,690	7,400		16,920	.00038	2,890	16,920	43,080	39.3	1.07
3	.240	.787	3,070	2,570	8,470		21,180	.00040	2,760	20,120	43,100	49.1	1.07
4	.241	.782	3,050	2,660	6,370		25,470	.00058	2,780	25,470	41,280	61.7	1.05
5	.241	.782	3,030	2,700	6,370	25,470	29,710	.00062	2,790	24,410	41,230	72.1	1.03
6	.243	.777	2,880	2,300	6,360	27,540	33,900	.00095	2,360	26,480	40,520	83.7	1.03
Average			3,020	2,600	6,870	26,500					42,070		
Laminate 143-114, Resin 2 -- Cross-laminated													
TCO-9-25-3	.235	.785	2,570	2,230	5,420	20,600	21,680	.00091	2,320	20,600	45,050	48.1	1.04
4	.237	.780	2,660	2,260	5,410	21,640	27,050	.00108	2,360	21,640	43,000	62.9	1.04
5	.236	.778	2,660	2,440	5,450	31,590	32,680	.00115	2,530	32,680	44,390	73.6	1.04
Average			2,630	2,310	5,430	24,610					44,150		
Laminate 112-114, Resin 2 -- Parallel-laminated													
TAO-9-9-3	.253	.780	2,490	2,300	7,090		15,200	.00048	2,430	15,200	36,180	42.0	1.06
5	.256	.781	2,300	2,090	7,000	19,010	21,010	.00088	2,160	19,010	35,610	59.0	1.03
Average			2,400	2,200	7,040	19,010					35,900		
Laminate 184-114, Resin 2 -- Parallel-laminated													
TNO-9-56-3	.237	.790	2,990	2,620	7,480	22,430	26,710	.00097	2,730	22,430	46,570	57.4	1.04

[1] Values based on initial straight portion of stress-strain curve.

[2] Values based on second straight portion of stress-strain curve.

[3] Values from the six dry tests are double those actually observed on a 1-inch gage length, in order to be on a basis comparable with the other tests.

TABLE 4 RESULTS OF COMPRESSION TESTS OF 181-114 LAMINATE AND 143-114 LAMINATE SHOWING EFFECT OF PRELOADING ON THE MECHANICAL PROPERTIES AND OBSERVED SET

Specimen No.	Thickness	Width	Number of preload runs	First run: Modulus of elasticity	First run: Proportional-limit stress	First run: Maximum applied stress	First run: Observed set in 2-inch gage length	Final run: Modulus of elasticity	Final run: Proportional-limit stress	Final run: Ultimate stress[1]	Ratio of maximum applied stress to ultimate stress
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
	Inch	Inch		1,000 p.s.i.	P.s.i.	P.s.i.	Inch	1,000 p.s.i.	P.s.i.	P.s.i.	Percent
181-114 Laminate -- Control Specimens											
CBO-9-54-2	0.117	0.994	....	3,590	28,380	41,190	....	....	....	41,190	100.0
54-5	.117	.995	....	3,632	25,770	38,140	....	....	....	38,140	100.0
54-8	.120	.995	....	3,570	22,610	41,370	....	....	....	41,370	100.0
54-11	.120	1.008	....	3,472	22,320	38,940	....	....	....	38,940	100.0
70-2	.126	.990	....	3,409	25,650	37,120	....	....	....	37,120	100.0
70-5	.131	.985	....	3,275	24,800	37,470	....	....	....	37,470	100.0
70-8	.131	.984	....	3,198	24,820	35,760	....	....	....	35,760	100.0
70-11	.133	1.010	....	3,079	23,820	36,700	....	....	....	36,700	100.0
Average...	....	....	....	3,403	24,770	38,340	....	....	....	38,340	100.0
181-114 Laminate -- Preloaded Specimens											
CBO-9-54-7	.121	.996	1	3,377	....	6,640	0.00010	3,377	20,740	36,510	18.2
70-3	.134	.992	1	3,009	....	9,030	.00003	3,009	21,060	37,070	24.4
70-10	.133	1.010	1	3,127	....	10,420	.00011	3,127	19,360	31,820	32.7
54-3	.121	.994	1	3,358	....	13,300	.00000	3,358	24,940	37,750	35.2
54-10	.120	1.008	1	3,379	....	14,880	.00013	3,379	29,760	37,860	39.3
70-4	.135	.990	1	3,169	....	14,960	.00016	3,169	17,960	37,560	39.8
54-12	.122	1.012	1	3,381	....	21,060	.00041	3,381	25,920	38,470	54.7
54-4	.123	.995	1	3,333	....	19,610	.00019	3,333	22,060	32,680	60.0
70-6	.135	.984	3	3,117	16,560	21,080	.00038	3,117	21,080	33,500	62.9
70-12	.133	1.009	1	3,178	17,140	22,360	.00030	3,178	19,370	33,760	66.2
70-1	.135	.991	1	3,078	20,930	26,910	.00043	3,078	22,420	37,070	72.6
54-6	.120	.996	1	3,443	19,240	26,770	.00041	3,443	24,260	36,480	73.4
54-9	.120	1.007	1	3,333	23,170	28,140	.00040	3,333	23,170	37,070	75.9
70-9	.133	1.009	1	3,190	17,880	28,320	.00053	3,190	23,850	32,940	86.0
70-7	.135	.984	1	3,121	18,070	30,110	.00054	3,121	22,580	34,180	88.1
Average...	....	....	....	3,240	19,000	....	....	3,240	22,570	35,650	....
143-114 Laminate -- Control Specimens											
CCO-9-49-2	.121	.988	....	5,302	26,770	39,310	....	....	....	39,310	100.0
49-5	.121	.994	....	5,218	19,950	46,060	....	....	....	46,060	100.0
49-8	.123	.996	....	5,352	25,120	43,100	....	....	....	43,100	100.0
49-11	.123	1.001	....	4,913	28,430	43,130	....	....	....	43,130	100.0
69-2	.121	.988	....	5,061	28,440	50,610	....	....	....	50,610	100.0
69-5	.122	.989	....	5,133	26,520	49,890	....	....	....	49,890	100.0
69-8	.121	.984	....	5,069	40,310	50,810	....	....	....	50,810	100.0
69-11	.122	.989	....	5,207	29,010	44,170	....	....	....	44,170	100.0
Average...	....	....	....	5,157	28,190	45,880	....	....	....	45,880	100.0
143-114 Laminate -- Preloaded Specimens											
CCO-9-69-9	.124	.993	1	5,148	....	10,560	.00002	5,148	25,990	39,960	26.4
49-6	.124	.995	1	5,237	....	12,160	.00010	5,237	17,830	43,280	28.1
49-9	.122	1.001	1	5,251	....	14,740	.00003	5,251	16,380	47,080	31.3
69-3	.124	.989	1	5,337	15,490	16,310	.00005	5,337	19,570	44,600	36.5
69-12	.123	.989	1	4,983	....	18,910	.00016	4,983	23,020	40,230	46.9
49-12	.124	1.002	1	5,337	17,710	22,540	.00010	5,337	19,320	40,240	56.0
69-7	.124	.989	3	5,124	25,260	26,910	.00018	5,124	30,930	43,630	61.7
49-4	.126	.990	1	5,137	24,050	28,060	.00016	5,137	28,860	43,610	64.3
69-4	.124	.988	1	4,986	17,960	24,490	.00027	4,986	26,120	37,710	64.9
69-6	.124	.988	1	5,129	27,750	32,650	.00012	5,129	31,020	44,240	73.8
49-7	.125	.995	1	5,259	25,730	30,550	.00018	5,259	30,550	40,440	75.5
49-1	.124	.991	1	4,949	27,670	36,620	.00022	4,949	26,040	42,800	85.6
69-1	.125	.990	1	5,090	16,160	40,400	.00033	5,090	29,090	42,670	94.7
49-10	.122	1.001	1	5,149	22,930	39,300	.00012	5,149	22,930	40,370	97.3
Average....	....	....	....	5,151	22,070	....	....	5,151	24,840	42,210	....

[1]The 0.2 percent offset yield stress is not reached before the ultimate stress.

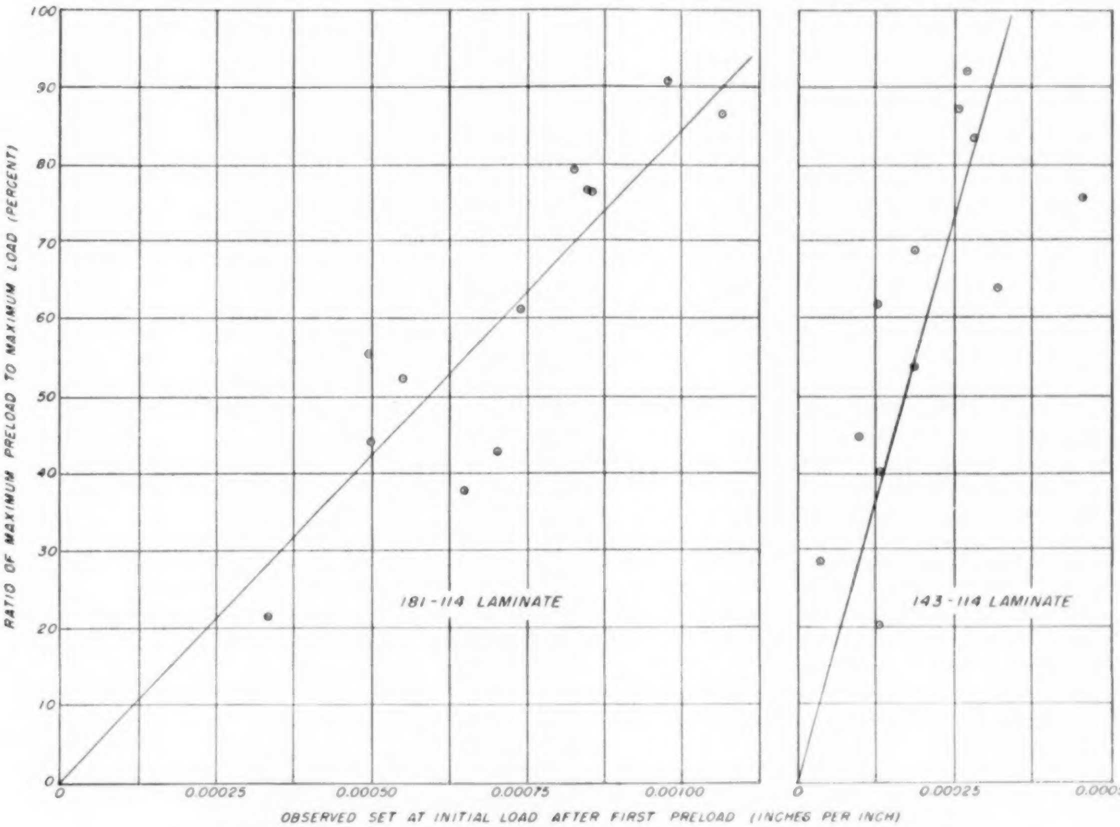

FIG. 4 THE EFFECT OF TENSILE PRELOADING ON OBSERVED SET OF TWO PLASTIC LAMINATES
(Observed set is elongation at initial load resulting from loading specimen to maximum preload a single time.)

when preloaded beyond the secondary proportional limit; the same laminate, tested wet, showed a difference of only about 3 per cent.

SUMMARY OF EFFECTS OF PRELOADING

The following summary is based largely on tests of a single laminate; limited check tests of other laminates generally confirm these observations:

1 The dual straight lines observed in the tensile load-deformation curves of many glass-fabric laminates are not representative of the load-deformation relationships of the laminate after it has been stressed beyond its initial proportional limit.

2 The tensile modulus of elasticity after preloading beyond the initial proportional limit is intermediate between the initial and secondary moduli commonly reported. The ratio of modulus of elasticity after preloading to the secondary modulus of elasticity varies with the degree of preloading, the modulus after preloading approaching the secondary modulus at high levels of preloading.

3 The stress at proportional limit (tension) after preloading corresponds with the preload up to a maximum equal to the secondary proportional limit.

4 Compressive properties are not affected appreciably by preloading.

5 The effect of preloading on laminates in the wet condition is similar to that found for laminates in the dry condition. The ratio of final modulus of elasticity to secondary modulus is somewhat smaller for wet than for dry laminates for the same degree of preloading.

6 Maximum tensile strength does not appear to be affected by preloading.

FATIGUE[3]

Materials Tested. Three laminates were tested. Two of the laminates were made with a satin-weave fabric (181-114) having essentially the same properties in the directions parallel and perpendicular to warp, while the other was made with a unidirectional fabric (143-114).

Laminate 1 consisted of 181-114 fabric parallel-laminated with a laminating resin of the polyester (alkyd-styrene) type.

Laminate 2 consisted of 181-114 fabric parallel-laminated with a laminating resin of the polyester (diallyl phthalate-alkyd) type.

Laminate 3 consisted of 143-114 fabric cross-laminated with the same resin that was used in laminate 1.

General characteristics of the laminates tested are shown in Table 5.

Methods of Test. Specimen types: The specimen used for fatigue tests was designed to be stable when loaded in compression even without lateral support. It was 6 in. long and dumbbell-shaped, the $^1/_2$-in-wide minimum section being connected to the

[3] "Fatigue Tests of Glass-Fabric-Base Plastic Laminates Subjected to Axial Loading," by K. H. Boller, Forest Products Laboratory Report No. 1823, 1951.

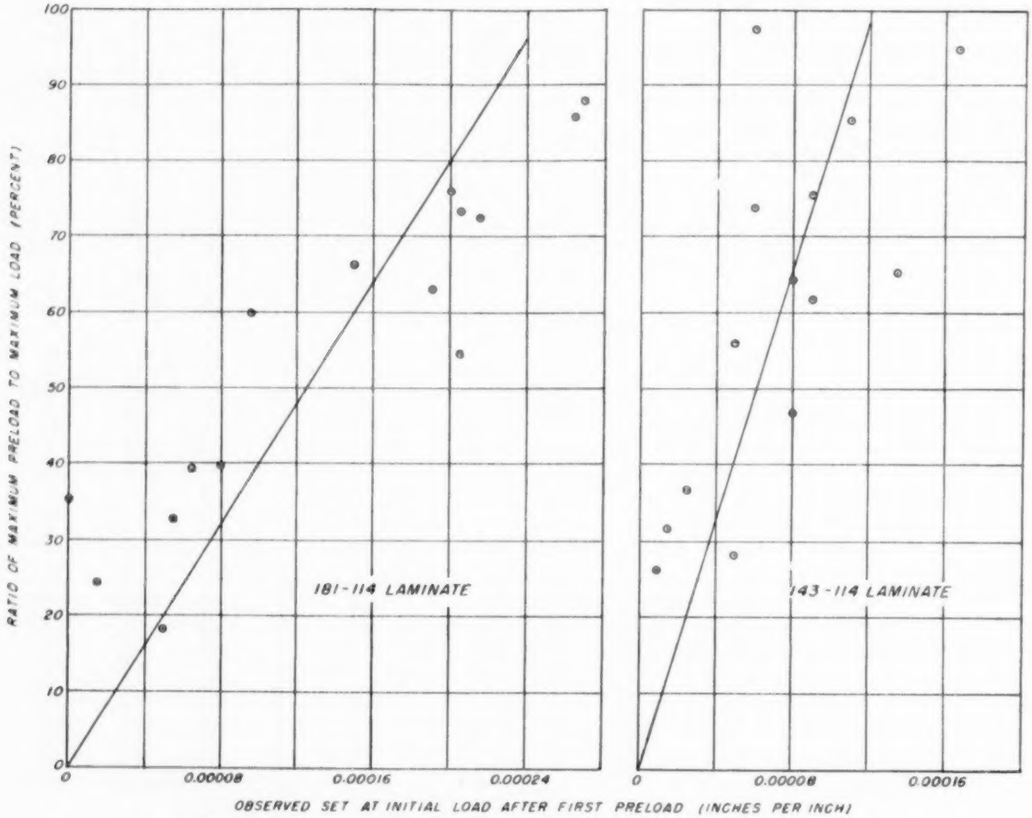

FIG. 5 EFFECT OF COMPRESSIVE PRELOADING ON OBSERVED SET OF TWO PLASTIC LAMINATES
(Observed set is contraction at initial load resulting from loading specimen to maximum preload a single time.)

TABLE 5 GENERAL PROPERTIES OF $^1/_4$-IN-THICK LAMINATES TESTED IN FATIGUE

Laminate number	Fabric	Direction of plies	Total number of plies	Specific gravity	Resin content, per cent	Barcol hardness
1	181-114	Parallel	23	1.76	39.0	69
2	181-114	Parallel	23	1.79	40.6	71
3	143-114	Cross	26	1.81	33.0	65

$1^1/_2$-in-wide ends by circular arcs of 4-in. radius. The specimen was clamped at each end so that only about $2^1/_4$ in. of the 6-in. length were unsupported. Since the specimens were $^1/_4$ in. thick, the ratio of unsupported length to thickness was about 9. Theoretical considerations[4] indicate the ratio of stress at the outside edge of the minimum section to the average stress to be about 1.1.

Some of the fatigue specimens contained a stress raiser in the form of a circular hole, $^1/_8$ in. diam, at the center of the width at the minimum section. When subjected to a direct stress along the natural axis, $\theta = 0$ deg, the theoretical stress at the edge of this hole is about $3^1/_2$ times the average stress for specimens loaded parallel to warp and for stresses within the proportional limit. This concentration is released somewhat by plastic flow when the stresses exceed the proportional limit.[5]

A few specimens were tested in bolt bearing. The fatigue specimen was modified by increasing the diameter of the hole from $^1/_8$ to $^1/_4$ in. and by removing 2 in. from one end, thus providing a length (end distance) of $^7/_8$ in. from the edge of the loading pin to the end of the specimen.

Static (control) tests in tension and compression were made on specimens of the types described in the section on preloading. Static tests were made also on specimens having the same form as that used in the fatigue tests, both with and without holes.

In general, the fatigue specimens failed at the net section. Only about 5 per cent of the specimens failed at the edge of the clamps.

Methods of test: All of the fatigue specimens were loaded axially in a direct-stress fatigue machine. The specimens were clamped at their ends by grips bolted to the loading screws of the testing machine, care being exercised to assure axial loads on the specimen. Load was applied at the rate of 900 cycles per min in a machine having a maximum capacity of 10,000 lb.

For the bolt-bearing tests, load was applied to one end of the specimen by means of the clamp used for the regular fatigue tests and to the $^1/_4$-in. steel pin near the other end by means of a steel frame attached to a loading screw.

The ratio of minimum to maximum load (stress ratio) was kept as nearly constant as possible by periodic checks and adjustments when necessary. An electronic shutoff mechanism stopped the test if the load dropped more than about 50 lb. If failure had not occurred when the machine stopped, adjustments were made and the test continued.

[4] "Effect of Hyperbolic Notches on the Stress Distribution in a Wood Plate," by C. B. Smith, *Quarterly of Applied Mathematics*, vol. 6, January, 1949, pp. 452-456.

[5] "Effect of Elliptic and Circular Holes on the Stress Distribution in Plates of Wood or Plywood Considered as Orthotropic Materials," by C. B. Smith, Forest Products Laboratory Report No. 1510, 1944.

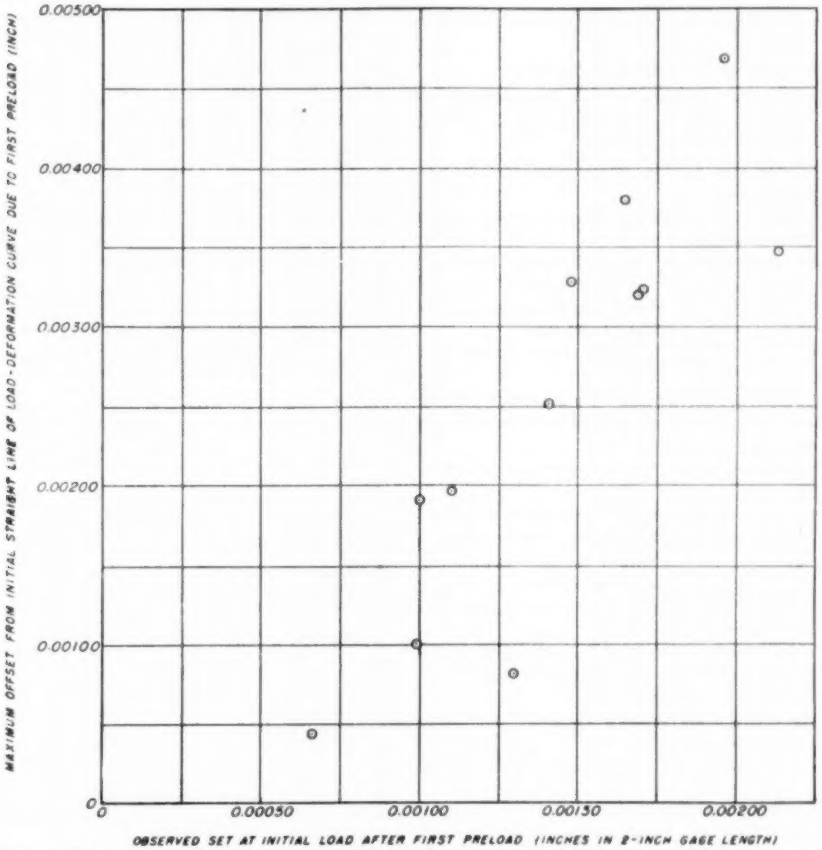

Fig. 6 Relationship Between Offset and Observed Set in Tension for 181-114 Laminate, as Observed From First Preload

(Values measured from load-deformation curve.)

Conditioning: Part of the specimens were conditioned at 75 F and 50 per cent relative humidity prior to test and were tested under these conditions. The majority of the specimens were continuously cooled during test by an electric fan. A few were tested without being cooled.

Some specimens were conditioned, prior to test, by exposure to 100 F and approximately 100 per cent relative humidity. High relative-humidity conditions during test were maintained by enclosing the specimen and the loading grips in a cylinder through which air at 75 F and 98 per cent relative humidity was passed.

Test variables: Nineteen series of fatigue tests involving a number of variables were made. A brief description of each

TABLE 6

SUMMARY OF CONDITIONS FOR FATIGUE TESTS OF GLASS-FABRIC LAMINATES

Test series	Laminate number	Mean stress[a]	Angle between load and warp direction, deg	Type of specimen: Unnotched	Type of specimen: Notched	Type of specimen: Bolt bearing	Exposure: 75 F 50 per cent relative humidity	Exposure: 75 F 98 per cent relative humidity	Moving air: Yes	Moving air: No
1	1	0	0	x			x		x	
2	1	0	0		x		x		x	
3	1	0	0	x			x			x
4	1	0	0	x				x	x	
5	1	0	0		x			x	x	
6	1	0	45	x			x		x	
7	1	0	45		x		x		x	
8	1	0	45	x				x	x	
9	1	0	45		x			x	x	
10	2	0	0	x			x		x	
11	2	0	0	x			x			x
12	2	0	0		x		x		x	
13	3	0	0	x			x		x	
14	3	0	0		x		x		x	
15	1	0	0			x	x		x	
16	1	1/2	0	x			x		x	
17	1	1/4	0	x			x		x	
18	1	1/2	0		x		x		x	
19	1	1/4	0		x		x		x	

[a] Expressed as a proportion of the static tensile strength.

series is given in the following, and test data are summarized in Table 6.

The series tested at zero mean stress were as follows:

1 Unnotched specimens from laminate 1, loaded at 0 deg to warp, and tested at 75 F and 50 per cent relative humidity while cooled by a blast of air.

2 Notched specimens ($^1/_8$ in. hole) from laminate 1, loaded and tested as under (1).

3 Unnotched specimens from laminate 1, loaded and tested as under (1) without the air cooling.

4 Unnotched specimens from laminate 1, exposed about 30 days to 100 F and 100 per cent relative humidity and then loaded at 0 deg to warp, and tested in a chamber having continuously moving air at 75 F and 98 per cent relative humidity.

5 Notched specimens from laminate 1, exposed, loaded, and tested as under (4).

6 Unnotched specimens from laminate 1, loaded at 45 deg to warp, and tested at 75 F and 50 per cent relative humidity while cooled by a blast of air.

7 Notched specimens from laminate 1, loaded and tested as under (6).

8 Unnotched specimens from laminate 1, exposed about 30 days to 100 F and 100 per cent relative humidity and then loaded at 45 deg to warp, and tested in a chamber having continuously moving air at 75 F and 98 per cent relative humidity.

9 Notched specimens from laminate 1, exposed, loaded, and tested as under (8).

10 Unnotched specimens from laminate 2, loaded at 0 deg to warp, and tested at 75 F and 50 per cent relative humidity while cooled by a blast of air.

11 Unnotched specimens from laminate 2, loaded and tested as under (10) without the air cooling.

12 Notched specimens from laminate 2, loaded and tested as under (10).

13 Unnotched specimens from laminate 3, loaded at 0 deg to warp, and tested at 75 F and 50 per cent relative humidity while cooled by a blast of air.

14 Notched specimens from laminate 3, loaded and tested as under (13).

15 Notched specimens from laminate 1, loaded at 0 deg to warp through a pin in the notch and tested at 75 F and 50 per cent relative humidity while cooled by a blast of air. The notch and pin in this series were $^1/_4$ in. diam.

The following series of specimens were tested with mean stress other than zero:

16 Unnotched specimens from laminate 1, loaded at 0 deg to warp, and tested at 75 F and 50 per cent relative humidity while cooled by a blast of air. The mean stress used in testing was about one half of the ultimate tensile strength.

17 Unnotched specimens from laminate 1, loaded and tested as under (16), except with a mean stress of about one fifth of the ultimate tensile strength.

18 Notched specimens from laminate 1, loaded and tested as under (16), except with a mean stress of about one half of the notched ultimate tensile strength.

19 Notched specimens from laminate 1, loaded and tested as under (16), except with a mean stress of about one fifth of the notched ultimate tensile strength.

Presentation and Discussion of Data. In the succeeding discussion, fatigue ratios are always expressed as percentages of the static tensile strength of an unnotched specimen, tested at 75 F and 50 per cent relative humidity.

Fatigue strength of laminates: The fatigue-strength values of unnotched specimens of laminates 1 and 3, tested parallel to warp, were similar, both laminates having a fatigue strength, at 10 million cycles, of about one fourth of the static tensile strength, Table 7 and Fig. 7. Laminate 2, however, made with a different resin, showed consistently higher fatigue-strength values in the lower stress ranges; they were between one fourth and one third of the static strength at 10 million cycles.

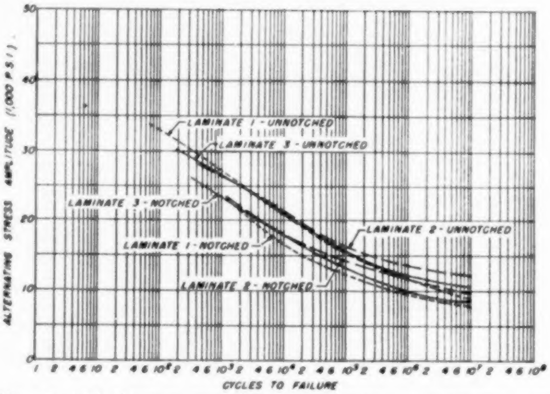

FIG. 7 S-N CURVES FOR THREE LAMINATES, UNNOTCHED AND NOTCHED, TESTED AT 75 F AND 50 PER CENT RELATIVE HUMIDITY WITH MEAN STRESS OF ZERO

(S-N, stress-number of cycles to failure.)

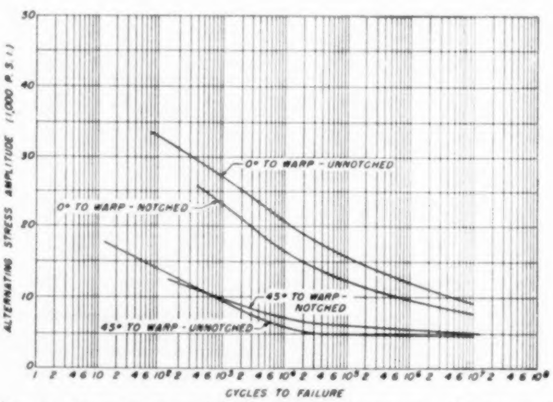

FIG. 8 COMPARISON OF S-N CURVES OF LAMINATE 1 TESTED AT 0 DEG AND 45 DEG TO WARP

The same general relationships among laminates exist also for notched specimens. The fatigue-strength values, however, are somewhat lower than for the unnotched specimens.

Effect of direction of loading: Since the static-strength values are markedly lower at 45 deg than at 0 deg to warp, so also are the fatigue-strength values for laminate 1, Fig. 8. This difference becomes smaller at the larger number of cycles, since the specimens tested at 45 deg reached an approximately constant fatigue strength (endurance limit) at about 40,000 cycles, while those tested at 0 deg to warp did not exhibit an endurance limit even at 10 million cycles.

Fatigue-strength values, expressed as percentages of the corresponding static strength, were not greatly different at 10 million cycles for specimens loaded at 0 deg and 45 deg to warp and tested dry, regardless of the direction of loading. When specimens were tested wet, however, fatigue strength at 10 million cycles was a lower proportion of the static strength at 45 deg than at 0 deg, Table 7.

For most specimens, fatigue ratios are generally on the order of 10 to 20 per cent lower for the notched than for the unnotched

TABLE 7 EFFECT OF VARIOUS TEST CONDITIONS ON THE FATIGUE STRENGTH OF GLASS-FABRIC LAMINATES TESTED IN AXIAL LOADING WITH ZERO MEAN STRESS

Test condition	Static tests				Fatigue tests					
	Tension		Compression		10,000 cycles			10,000,000 cycles		
	Strength	Reduction factor[1]	Strength	Reduction factor[1]	Fatigue strength	Reduction factor[1]	Fatigue ratio[2]	Fatigue strength	Reduction factor[1]	Fatigue ratio[2]
	P.s.i.		P.s.i.		P.s.i.			P.s.i.		
Laminate 1, Tested at 0° to Warp										
75° F., 50 percent relative humidity, unnotched	40,300	....	40,700	....	20,600	....	0.51	9,000	....	0.22
75° F., 50 percent relative humidity, notched	31,070	1.30	45,050	0.90	16,500	1.25	.41	7,500	1.20	.19
75° F., 98 percent relative humidity, unnotched	35,170	1.15	28,280	1.44	13,100	1.57	.33	9,000	1.00	.22
75° F., 98 percent relative humidity, notched	30,150	1.34	18,250	2.23	14,000	1.47	.35	7,750	1.16	.19
75° F., 50 percent relative humidity, unnotched, uncooled	40,300	1.00	40,700	1.00	18,000	1.14	.45	9,000	1.00	.22
75° F., 50 percent relative humidity, loaded through notch (bolt bearing)	....	....	....	....	16,500	1.25	.41	7,500	1.20	.19
Laminate 1, Tested at 45° to Warp										
75° F., 50 percent relative humidity, unnotched	22,010	....	23,060	....	5,760	....	.26	4,750	....	.22
75° F., 50 percent relative humidity, notched	18,960	1.16	24,440	.94	6,750	.85	.31	4,750	1.00	.22
75° F., 98 percent relative humidity, unnotched	13,870	1.59	14,410	1.60	3,870	1.49	.18	2,700	1.76	.12
75° F., 98 percent relative humidity, notched	13,550	1.62	15,560	1.48	4,250	1.35	.19	3,150	1.51	.14
Laminate 2, Tested at 0° to Warp										
75° F., 50 percent relative humidity, unnotched	41,220	....	34,220	....	21,220	....	.51	12,100	....	.29
75° F., 50 percent relative humidity, notched	31,510	1.31	38,590	.89	18,000	1.18	.44	9,500	1.27	.23
75° F., 50 percent relative humidity, unnotched, uncooled	41,220	1.00	34,220	1.00	18,000	1.18	.44	12,000	1.01	.29
Laminate 3, Tested at 0° to Warp										
75° F., 50 percent relative humidity, unnotched	42,540	....	43,050	....	20,700	....	.49	10,500	....	.25
75° F., 50 percent relative humidity, notched	38,820	1.10	44,800	.96	17,800	1.16	.42	7,800	1.34	.18

[1]Reduction factor is the ratio, at a given number of cycles, of the strength of unnotched material tested at 75° F. and 50 percent relative humidity to the strength of the material at some other test condition.

[2]Fatigue ratio is the ratio of the fatigue strength of a material at some test condition to the static tensile strength of the same material when unnotched and tested at 75° F. and 50 percent relative humidity.

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specimens. Fatigue-strength values, therefore, are lower for the notched than for the unnotched specimens. For laminates 1 and 2 at 0 deg to warp, however, the ratio of the fatigue strength of the notched specimen at a given number of cycles to that of the unnotched specimen at the same number of cycles is somewhat higher than is the corresponding ratio of static tensile-strength values. This indicates the possibility that plastic flow with an increasing number of stress reversals may reduce the stress-concentration effect. The reverse, however, is true of laminate 3, which is made with a different fabric.

Irregular results were obtained in some cases. Although methods are not available for evaluating the effect of stress concentration other than in the directions of the natural axes, it is obvious that stress-concentration effects must exist in those specimens tested at 45 deg. Both fatigue tests and static tests in compression at 45 deg, however, show higher strength values, based on the net section, for the notched than for the unnotched specimens, Table 7, Fig. 8. No explanation is offered for these irregular results, and the data are presented only to illustrate general trends.

Effect of method of applying load: In one series of specimens, load was applied through a pin inserted in a hole at the minimum section. Essentially the same fatigue results were found from this series as from the tests in which fatigue loads were applied at the ends of the specimens. This is evidenced by the fatigue-strength values at 10,000 and 10 million cycles shown in Table 7. The small number of tests involved precludes any definite conclusions as to whether the agreement in results between the two methods of tests is significant or is a coincidence.

Effect of moisture: Laminate 1 was tested after both dry and wet conditioning. Figs. 9 and 10 illustrate the effect of moisture on fatigue strength. At 0 deg to warp, it may be noted that the

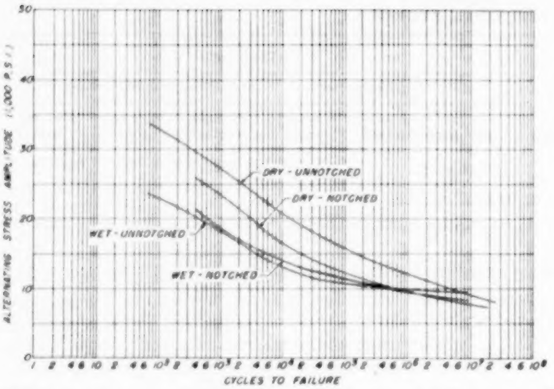

FIG. 9 COMPARISON OF S-N CURVES OF LAMINATE 1 TESTED AT 0 DEG TO WARP AFTER DRY AND WET CONDITIONING

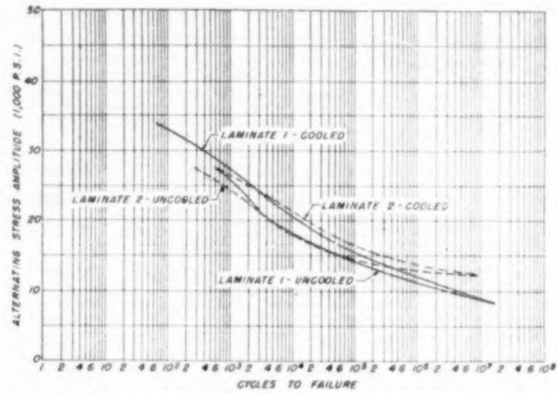

FIG. 11 COMPARISON OF S-N CURVES OF UNNOTCHED SPECIMENS OF LAMINATES 1 AND 2 TESTED WITH AND WITHOUT BEING COOLED

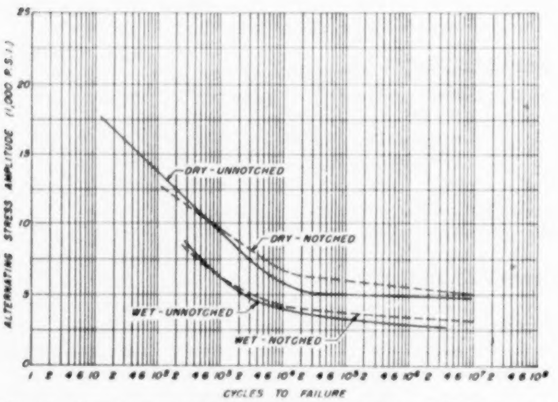

FIG. 10 COMPARISON OF S-N CURVES OF LAMINATE 1 TESTED AT 45 DEG TO WARP AFTER DRY AND WET CONDITIONING

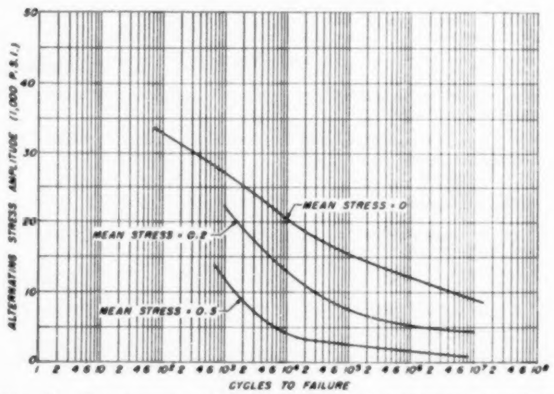

FIG. 12 COMPARISON OF S-N CURVES FOR UNNOTCHED SPECIMENS OF LAMINATE 1 WHEN TESTED AT MEAN STRESSES OF ZERO AND OF $^1/_5$ AND $^1/_2$ OF MEAN STATIC TENSILE STRENGTH

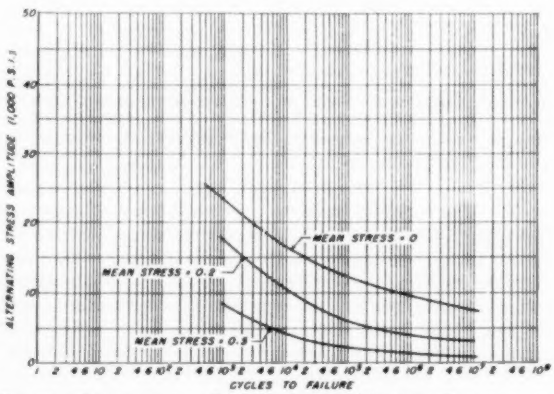

FIG. 13 COMPARISON OF S-N CURVES FOR NOTCHED SPECIMENS OF LAMINATE 1 WHEN TESTED AT MEAN STRESSES OF ZERO AND OF $^1/_5$ AND $^1/_2$ OF MEAN STATIC TENSILE STRENGTH

dry and wet fatigue-strength values tend to approach the same value as the number of cycles required to cause failure increases. It is probable that in spite of the high relative humidity maintained in the test chamber, sufficient heat is generated in the specimen, after a large number of stress reversals, to cause some drying and thus some increase in strength.

The same tendency was not present when the specimens were loaded at 45 deg to the warp. The reason for this difference is not known, but it may be related to the lesser generation of heat at the lower stress levels for the material loaded at 45 deg.

No consistent relation between fatigue-strength values for wet and dry specimens is evidenced by the data.

Effect of cooling during test: A few specimens of laminates 1 and 2 were tested without the use of the cooling fan. Temperature measurements at a stress level of 14,500 psi indicated a temperature rise of about 10 F during most of the test. Just prior to failure, however, the rise was about 70 F.

Fig. 11 indicates that the heat generated at the higher stress levels is sufficient to cause some deterioration in fatigue strength. The period at the highest stress levels, however, seems to be of sufficiently short duration that the effect of the heat does not become apparent. At the lower stress levels, apparently, the heat generated is not sufficient to cause appreciable deterioration, in spite of the longer period at these levels.

Effect of variation in level of mean stress: Figs. 12 and 13 compare the S-N (stress-number of cycles to failure) curves for specimens of laminate 1 tested at zero mean stress and at mean stresses of one fifth and one half of the static tensile strength. These data are summarized in Table 8 and in Figs. 14 and 15. The latter figures, obtained from the S-N curves of Figs. 12 and 13, permit the determination of fatigue strength, at any value of

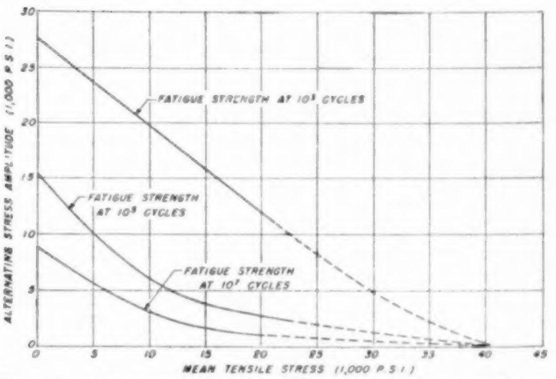

FIG. 14 ALTERNATING STRESS AMPLITUDES REQUIRED TO CAUSE FAILURE OF UNNOTCHED SPECIMENS OF LAMINATE 1 AT VARIOUS LEVELS OF MEAN STRESS

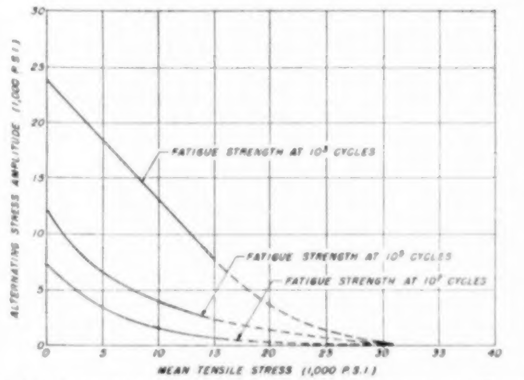

FIG. 15 ALTERNATING STRESS AMPLITUDES REQUIRED TO CAUSE FAILURE OF NOTCHED SPECIMENS OF LAMINATE 1 AT VARIOUS LEVELS OF MEAN STRESS

TABLE 8 EFFECT OF MEAN STRESS ON AXIAL FATIGUE STRENGTH OF LAMINATE 1 TESTED AT 75 F AND 50 PER CENT RELATIVE HUMIDITY

Mean stress[a]	Static tensile strength	Fatigue strength at 1000 cycles	Fatigue strength at 100000 cycles	Fatigue strength at 10000000 cycles
		UNNOTCHED		
0	40300	27200	15500	9000
1/5	40300	21700	7500	4200
1/2	40300	11500	2600	800
		NOTCHED		
0	31070	23300	12300	7500
1/5	31070	17200	5800	3000
1/2	31070	8200	2100	700

[a] Expressed as a proportion of the static tensile strength.

mean stress between zero and tensile strength, for three numbers of cycles of reversal to failure.

At 10 million cycles, the fatigue-strength values corresponding to a mean stress of one fifth of the static tensile strength were less than one half of the fatigue strength corresponding to a mean stress of zero. At a mean stress of one half of the static tensile strength, the corresponding ratio of fatigue strengths was only about one tenth. The fatigue strength at 10 million cycles is therefore only about 2 per cent of the static tensile strength when the mean stress is one half of the static tensile strength.

SUMMARY OF FATIGUE DATA

The following summary is based on tests of only three glass-fabric laminates and should be considered with this limitation in mind:

1 None of the laminates tested parallel to the warp direction exhibited an endurance limit. When loaded at 45 deg to the warp direction, however, fatigue-strength values were approximately constant after about 40,000 cycles.

2 Fatigue strength at 10 million cycles, for specimens tested in the dry condition, was in the range of 22 to 29 per cent of the static tensile strength.

3 The presence of a notch, in the form of a 1/8-in. hole, reduced the fatigue strength by about 10 to 20 per cent.

4 Specimens tested parallel to warp and under wet conditions exhibited about the same fatigue strength at 10 million cycles as did those tested under dry conditions. For smaller numbers of reversals to failure, those tested under wet conditions exhibited lower fatigue-strength values. When loaded at 45 deg to warp, however, the specimens tested under wet conditions showed a deficiency in fatigue strength compared to those tested in the dry condition, and this deficiency was approximately constant over the range tested.

5 When loaded through a pin inserted in a hole at the minimum section, the one laminate tested in this fashion exhibited the same fatigue characteristics as did notched specimens of the same laminate when loaded at their ends.

6 No cooling during test resulted in a lower fatigue strength, except near 10 million cycles where fatigue-strength values were about the same for both cooled and uncooled specimens.

7 As the mean stress is raised from zero to some tensile value, the amplitude of alternating stress that can be sustained at any number of cycles is sharply reduced. For example, at a mean stress of one half of the tensile strength, the amplitude required to cause failure at 10 million cycles is only about one tenth of the amplitude required at a mean stress of zero.

Studies of Submergence Requirements of High-Specific-Speed Pumps

By H. W. Iversen,[1] Berkeley, Calif.

Submergence requirements of high-specific-speed pumps that take suction from open sumps depend upon vortex formation in the sump and resulting air entrainment in the pumped liquid. Two interrelated requirements must be met in sump design. The sump must be of minimum size without affecting adversely the pump performance with respect to efficiency and head. Laboratory studies have been made on the effect of side-wall and bottom clearances, relative to the suction bell, on the pump efficiency, and also upon the submergence requirements to avoid vortex formation with air entrainment. Results show that side-wall and end-wall clearance of one quarter to one half the suction-bell diameter, and bottom clearances of one half the suction-bell diameter will not affect the normal pump performance adversely. Minimum stable depths of sump liquid also were obtained with these spacings. Model tests according to Froude's modulus do not yield quantitative results since frictional effects apparently influence the results. However, qualitative studies may be made with models to determine optimum sump configurations to avoid vortex formation.

Introduction

Deep-well turbine and propeller pumps installed in open sumps, in addition to head, capacity, and power ratings, require an added rating of submergence. The free water level in the sump must be sufficiently above the lip of the suction bell to permit proper pump operation. The submergence requirement is dictated either by the cavitation characteristics of the pump, or by the tendency for vortex formation in the sump with consequent air entrainment in the pumped liquid. The present discussion deals with the latter of these phenomena, that of air entrainment.

Economical sump design should result in a minimum sump volume and a minimum boundary area without affecting the pump performance adversely. The problem is then one of determining the minimum depth of water permissible in a sump, and the minimum side-wall clearances surrounding the pump column consistent with specified head and efficiency performance.

Information in the literature is scarce. Richardson[2] shows a recommended submergence curve in terms of inlet velocities at the suction bell of diameter D, for pumps positioned $D/2$ from the sump bottom, and with $D/2$ side clearances. Stepanoff[3] states the same general clearances without specifying values of submergence. Kerr and Moyer,[4] on the model tests of the condenser-cooling-water pumps of the Southwark Station, observed optimum pump efficiencies with a bottom clearance of approximately $D/2$. The available data are inconclusive on the complete aspects of the problem.

Tests have been made at the University of California to establish criteria for sump design. Three groups of tests were made, which can be classified as follows:

1 Effect of sump configuration on pump efficiency.
2 Effect of sump configuration on minimum submergence.
3 Possibilities of model studies.

In all tests the sump configuration consisted of vertical side walls bounding a rectangular plan-form horizontal bottom. The pump was located at one end of the sump. Inflow to the pump was parallel to the side walls. Fig. 1 shows a diagram of the sump arrangement.

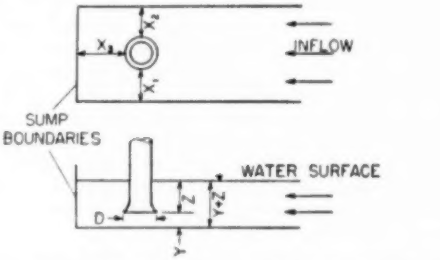

Fig. 1 Diagram of Sump Arrangement Showing Nomenclature

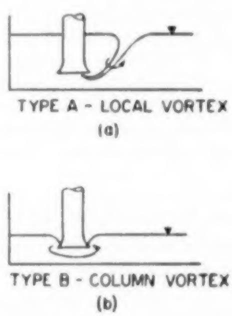

Fig. 2 Types of Vortex Formations

General Observations

Before reporting on the results of the test, some observations should be made on the flow patterns in the sump with respect to vortex formation and air entrainment.

The condition of a vortex formation is illustrated in Fig. 2. Two types of vortexes may be established. In Fig. 2(a) is shown a local vortex of a rotating region of liquid with the liquid surface

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[1] Associate Professor of Mechanical Engineering, University of California. Jun. ASME.

[2] "Submergence and Spacing of Suction Bells," by C. A. Richardson, Water Works and Sewerage, Reference and Data, Part 1, Water Supply, 1941, p. 25.

[3] "Centrifugal and Axial Flow Pumps," by A. J. Stepanoff, John Wiley & Sons, Inc., New York, N. Y., 1948, pp. 363–366.

Contributed by the Hydraulic Division and the Engineering Institute of Canada and presented at the Semi-Annual Meeting, Toronto, Ont., Can., June 11–14, 1951, of The American Society of Mechanical Engineers.

[4] "Hydraulic-Engineering Problems at Southwark Generating Station," by S. L. Kerr and S. Moyer, Trans. ASME, vol. 64, 1942, p. 539.

drawn down in a localized area. An air core is present that may be pulled into the pump suction if the vortex strength and inflow velocity are sufficient. Two effects, other than air entering the pump, may result. If the vortex is unstable, forms and breaks, then re-forms, the periodic air flow produces velocity fluctuations with a corresponding fluctuation in the impeller thrust. This leads to "bouncing" of the impeller and shaft assembly with possible serious effects on bearings and supports. The disturbance also affects the pump hydraulic efficiency. If the vortex is stable the bouncing may not be evident, but the hydraulic efficiency may be affected.

Fig. 2(*b*) diagrams a second condition that may be produced. The liquid "swirls" around the pump column with drawdown around the periphery of the column. The effects on the pump are the same as those mentioned for the local vortex.

What produces the vortex? A cursory examination would lead to the conclusion, since a vortex is a rotating column of fluid, that prerotation of the fluid entering the suction bell as a result of impeller action is the predominating influence. While this effect undoubtedly is present, the major influence stems, not from the pump suction, but from the sump design and the flow pattern in the sump. In some sumps a vortex is initiated far from the suction bell where any prerotation in the region of the suction bell could not be conceivably the cause.

A vortex is initiated from an eddy in the sump. When the eddy reaches the higher-velocity regions of flow near the suction bell, the lower portion of the eddy is constricted and drawn into the suction. Rotational velocities in the lower portion of the eddy are increased by the conservation of angular momentum, pressures are decreased, and an air core is established, Fig. 2. Hence any configuration of the sump boundaries at which an eddy forms is a potential source of a vortex. Sharp corners, piers, and irregular velocity distributions fall in this category. One unavoidable source of disturbance is the pump column. Since the column cannot be eliminated, sump design must take such a form that the column effect is minimized consistent with reasonable submergences.

As can be inferred, a complete study of all possible sump configurations to note the flow details, eddy formation, eddy growth into vortexes, air entrainment, efficiency effects, is an unlimited task. However, from general knowledge of flow phenomena, possible eddy-forming regions due to boundary configurations can be avoided.

The present tests were conducted on a simple rectangular sump configuration with a uniform inflow velocity as an approach to an optimum non-eddy-forming configuration. The general layout of the sump, with side and bottom-clearance designations, is shown in Fig. 1.

Effect of Sump Configuration on Pump Efficiency[5]

This series of tests were made with a vertical-shaft propeller pump rated at 4000 gpm and 20 ft head at 1160 rpm. The pump included a four-vane suction bell, short riser, and vaned discharge elbow. The suction-bell diameter was $20^1/_4$ in. Pump performance was determined by standard test procedures. Discharge was measured with standard ISA nozzles and water-mercury differential manometers, pressure heads with water or mercury columns, elevation heads with a transit and electric point gages, torque to pump shaft with a special dynamometer drive, and speed was controlled automatically at the rated value. The head of this unit was determined between the average water level adjacent to the pump and a point three pipe diameters downstream from the discharge elbow.

[5] "HP-14, Suction Conditions," by R. G. Folsom, Pump Testing Laboratory Technical Memorandum No. 6, University of California, December, 1940.

The normal performance of the pump, for comparative purposes, was obtained in a main test pit 8 ft × 8 ft × 20 ft deep. Performances with restricted side and bottom clearances were obtained in an 8-ft-deep × 4-ft-wide channel which connected the main reservoir with the main test pit. False walls of metal or wood were used to change the side-wall clearances. Performance was obtained over the range of flow rate in the region of maximum pump efficiency. The submergence was maintained constant.

Table 1 summarizes the test results. Runs 1 and 26 are for normal performance. All other runs are for restricted sumps as designated by dimensionless ratios of the side and bottom clearances compared to the suction-bell diameter. Over the range of flows from 3000 to 4500 gpm, it was found that the efficiency deviation for any one test as compared to the normal test results was a constant. The average efficiency deviation may be taken as a criterion of the effect of sump design on the performance. The deviation of the head, expressed as a per cent of the normal head, is also included in Table 1. In all tests there was no serious vortex formation with air entrainment.

Inspection of the efficiency comparisons shows best performance with side-wall and bottom clearances approximately one half the suction-bell diameter. Note that the head deviation is of the same order of magnitude as the efficiency deviation for any one run. The results also indicate that side-wall clearances of $D/4$ with a bottom clearance of $D/2$ do not affect the pump performance adversely.

Effect of Sump Boundaries on Minimum Stable Submergence

In order to ascertain critical submergence trends with variations in sump geometry, tests were made using a standard ASME nozzle shape on the end of a 4-in. (nominal) vertical pipe, Fig. 3. The outside diameter of the nozzle entrance was 6.375 in. The nozzle simulated a nonvane suction bell. Flow rates were measured by means of a calibrated ISA orifice in the pump-discharge line. An air separator was placed in the system between the nozzle and the pump. Two air separators also were placed in the discharge line of the pump. All separators were arranged to collect all the air that entered the nozzle to establish air-flow rates.

The nozzle was mounted in a large rectangular open tank. Side-wall partitions of $\frac{1}{2}$-in. plywood, 6 ft long, were used to form side clearances. The nozzle and suction line were connected to the permanent pump-suction line by a flexible hose to permit

TABLE 1 EFFECT OF SIDE AND BOTTOM CLEARANCES ON PUMP EFFICIENCY AND HEAD

Run	Bottom clearance, Y/D	Side-wall clearance, $X_1/D = X_2/D = X_3/D$	Submergence, Z/D	Maximum pump efficiency, per cent	Average efficiency difference (per cent of normal)	Average head difference (per cent of normal)
1	..	..	...	78.7	Normal	Normal
26	..	..	...	78.7	Normal	Normal
2	0.25	0.00	2.0	73.0	−8.3	−7.9
5	0.25	0.05	2.0	76.3	−3.2	−4.7
4	0.25	0.12	2.0	78.2	0.0	−0.1
3	0.25	0.24	2.0	78.7	0.0	−0.3
6	0.52	0.00	2.0	77.3	−1.9	−2.6
7	0.52	0.25	2.0	78.7	0.3	0.3
8	0.52	0.47	2.0	79.1	0.7	0.4

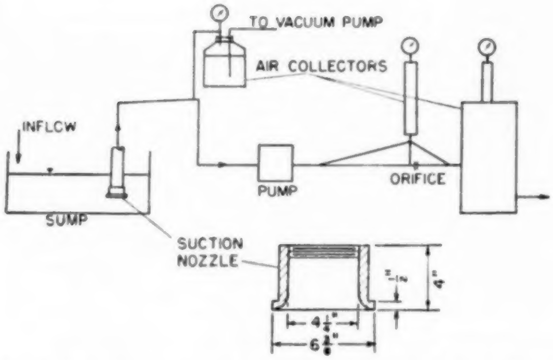


Fig. 3 Schematic Arrangement of Suction-Nozzle-Test Setup

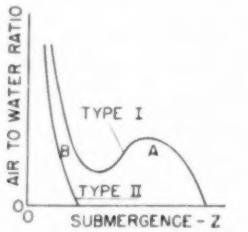


Fig. 4 General Nature of Air-to-Water Ratio as a Function of Submergence

(Region A corresponds to vortex Type A. Region B corresponds to vortex Type B.)

vertical and end positioning of the nozzle. The sump arrangement thus corresponded to Fig. 1.

With each sump geometry, the inflow to the sump tank from the laboratory supply was adjusted to equal the outflow through the nozzle. The water level was dropped in increments and, at each depth, flow rates of air and water were measured. Air quantities were reduced to atmospheric volumes. Observations also were made of the flow pattern in the sump in the vicinity of the nozzle.

In general, the air-to-water-rate ratio, expressed as a percentage, as a function of submergence resulted in two patterns, Fig. 4. The Type I relationship resulted from a series of events in the flow pattern. At large submergences eddies in the region of the pump column were slow-moving. As the submergence decreased, the eddies became stronger with a tendency for vortex formation. At a critical submergence a vortex was formed with an air core that reached the nozzle inlet, Fig. 2(*a*). These phenomena usually were unstable, a vortex would form, then break, and a new vortex form and break. As the submergence was lowered below the critical value the vortex formation became more stable with consequent greater air rates into the nozzle. A second critical submergence was reached when the inflow started to swirl around the column. Air rates usually decreased (with some sump configurations to zero); the vortex-formation tendency in Fig. 2(*a*) decreased. At a lower submergence the flow pattern developed into that of Fig. 2(*b*), a drawdown surrounding the column, with rapid rotation of the water around the column. Air entered the nozzle intermittently with a considerable interruption effect on the water flow.

Type II air rates as a function of submergence corresponded to the drawdown and rotation condition of Type I. No significant local vortex was observed.

Results of submergences at which air first entered the nozzle as a function of geometrical arrangement are given in Table 2. Fig. 5 shows the trends of critical submergence for a constant flow rate of 220 gpm. For symmetrical side and end-wall clearances, minimum submergences occur with close side-wall clearances and large bottom clearances. However, minimum depths in the sump occur with close side-wall clearances and small bottom clearances.

Certain advantages are noted with a nonsymmetrical location of the suction nozzle with respect to the walls, Table 2. As the end-wall clearance is reduced to $D/4$, or less, the critical submergence is reduced with no consistent variation with respect to side clearance. A zero end-wall clearance, with any tested side-wall clearance, resulted in Type II flow patterns and no noticeable local vortex formation in the sump except with large bottom clearance.

Table 3 lists some critical submergences with different flow rates. As is expected, the critical submergence becomes larger as the flow rate increases. The eddy and vortex pattern in the sump becomes more pronounced in the direction of the Type I flow pattern as the flows, and hence sump velocities, increase.

TABLE 2 RESULTS OF SUCTION-NOZZLE TESTS FOR CRITICAL SUBMERGENCES; CONSTANT FLOW RATE OF 220 GPM

Run	Side-wall clearances X_1/D	Side-wall clearances X_2/D	End-wall clearance, X_3/D	Bottom clearance, Y/D	Critical submergence, Z/D	Critical depth, $\frac{Z+Y}{D}$	Air-flow type (Fig. 4)
53	0.5	0.5	0.5	0.5	1.18	1.68	I
31	0.5	0.5	0.5	1.0	0.96	1.96	I
68	0.5	0.5	0.5	2.0	0.73	2.73	I
50	1.0	1.0	1.0	0.5	1.29	1.79	I
30	1.0	1.0	1.0	1.0	1.07	2.07	I
63	1.0	1.0	1.0	2.0	1.11	3.11	I
45	2.0	2.0	2.0	0.5	2.52	3.02	I
37	2.0	2.0	2.0	1.0	1.63	2.63	I
60	2.0	2.0	2.0	2.0	1.39	3.39	I
54	0.5	0.5	0.0	0.5	0.22	0.72	II
32	0.5	0.5	0.0	1.0	0.20	1.20	II
66	0.5	0.5	0.0	2.0	0.54	2.54	I
51	1.0	1.0	0.0	0.5	0.25	0.75	II
35	1.0	1.0	0.0	1.0	0.32	1.32	II
65	1.0	1.0	0.0	2.0	0.26	2.26	I
49	2.0	2.0	0.0	0.5	0.25	0.75	II
36	2.0	2.0	0.0	1.0	0.1	1.10	II
70	2.0	2.0	0.0	2.0	0.20	2.20	II
55	0.5	0.5	0.25	0.5	0.44	0.94	II
57	0.5	0.5	0.25	1.0	0.60	1.60	I
67	0.5	0.5	0.25	2.0	0.54	2.54	I

TABLE 3 RESULTS OF SUCTION-NOZZLE TESTS FOR CRITICAL SUBMERGENCE WITH VARIABLE FLOW RATES

Run	Flow rate, gpm	Side-wall clearances X_1/D	Side-wall clearances X_2/D	End-wall clearance, X_3/D	Bottom clearance, Y/D	Critical submergence, Z/D	Air flow type (Fig. 4)
55	220	0.5	0.5	0.25	0.5	0.44	II
77	350	0.5	0.5	0.25	0.5	0.98	I
78	410	0.5	0.5	0.25	0.5	1.16	I
67	220	0.5	0.5	0.25	2.0	0.54	I
76	350	0.5	0.5	0.25	2.0	1.29	I

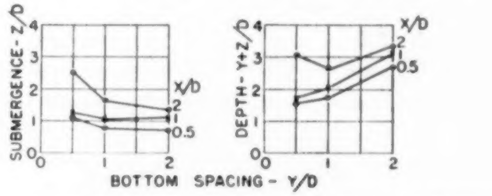


Fig. 5 Critical Submergence for Various Side and Bottom Clearances With Symmetrical Pump Location, $X_1 = X_2 = X_3 = X$

(Nozzle diameter = 6.375 in.; flow rate = 220 gpm.)

Model Studies

The question arises as to whether the reported tests with the nozzle are valid to show submergence phenomena with actual pumps. The force fields in the sump flow, with vortex formation,

include pressure, inertia, gravity, viscosity, and surface tension. As a first approximation, surface tension and viscous forces may be assumed small as compared to gravity forces. Hence Froude's modulus should apply with these restrictions

$$\text{Froude's modulus} = \frac{V}{\sqrt{Lg}}$$

where V = velocity, L = length, g = gravity constant.

Since g is constant, the application of Froude's modulus results in velocities proportional to the square root of the length. With geometrically similar systems, since flow rate = $Q = VA$, where A = area αL^2 then $Q \alpha L^{5/2}$.

An attempt was made to compare the results of the nozzle tests with a prototype pump. A propeller pump with a suction-bell diameter of 18.25 in., rated at 2900 gpm and 12 ft head at 370 rpm, was obtained and set up in the test basin described for the restricted-clearance efficiency tests. The pump discharged to a closed vertical cylindrical tank and thence to a metering section, Fig. 6. The tank acted as an air separator to permit measurement of air rates. After installation with available elevations and piping, it was found that the maximum delivery from the pump was approximately 2600 gpm.

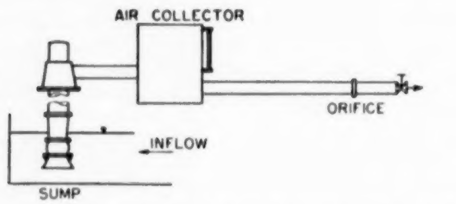
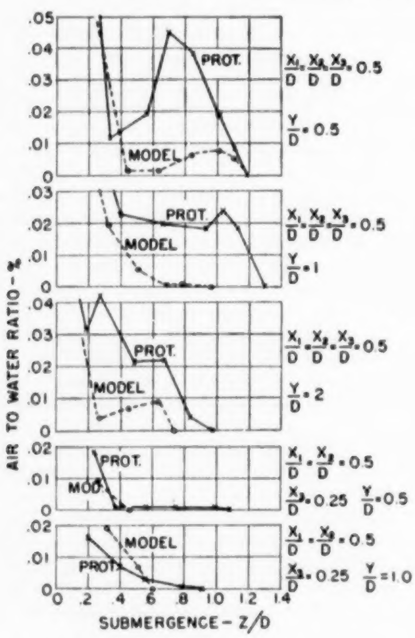


FIG. 6 DIAGRAM OF PROTOTYPE-PUMP SUBMERGENCE-TEST SETUP



Prototype: D = 18.25 in. Flow = 2600 gpm
Model: D = 6.375 in. Flow = 220 gpm

FIG. 7 COMPARISON OF MODEL AND PROTOTYPE AIR RATES AS A FUNCTION OF SUBMERGENCE

Runs with the 6.375-in. nozzle, for comparative sump geometries that could be installed in the prototype channel, were made with 220 gpm. The length ratio, based upon the suction-bell diameters, was 18.25/6.375, or 2.85. The prototype flow rate, corresponding to the model flow of 220 gpm, based upon $Q \alpha L^{5/2}$ should have been 3000 gpm. Thus a strict model-prototype situation was not realized. However, certain relationships become apparent when results of model and prototype are compared even though a discrepancy in flow was present. Percentage air rates are plotted in Fig. 7 as a function of submergences for both the prototype and the model. In general, the trends of air flows are similar. However, the absolute values differ, with smaller air rates and lower critical submergences in the model. The expectation would be the opposite since the prototype flow rate was smaller than that evaluated from the model law.

The evidence indicates that either surface tension or frictional forces are not negligible. Surface curvatures were not measured in the model, but observations of the vortex shapes, with estimates of differential pressures due to surface tension and curvatures, lead to the conclusion that this effect is negligible. For frictional similarity, Reynolds modulus should be satisfied. With the same fluid, water, in the model and prototype, Reynolds modulus results in velocities inversely proportional to the length. To satisfy this requirement the prototype flow should be considerably lower than those predicted from a consideration of Froude's modulus. Thus the frictional effects act in the manner shown in the results.

With more comparisons of this nature, a valid model law might be established. The indication is that the flow-rate ratio is equal to the length ratio to some exponent less than 2.5. However, a qualitative comparison between model and prototype does exist as is shown in Fig. 7. The conclusions as obtained from the nozzle tests are valid in essence if not in absolute magnitude.

SUMMARY

Unfortunately, as is very often the situation, further tests will have to be made to establish definite results for submergence requirements even for the simple sump geometry as investigated in the present tests. The following indications result from the three series of tests:

1 Side-wall and end-wall clearances in the range of $D/2$ to $D/4$ will not affect normal pump performance of head and efficiency adversely.

2 Bottom clearance should be approximately $D/2$ for negligible effect on the pump performance and for minimum depth of flow as determined by critical submergence.

3 Model tests based upon Froude's modulus will not yield quantitative results on submergence limits but will yield qualitative results from observation of flow patterns and eddy-forming regions in a sump.

4 Model tests relative to (3) may be made with suction nozzles and do not require a model pump.

Further work, particularly on efficiency and head comparisons, should be made with pumps in the reported sump configuration with end-wall clearances of $D/4$ to zero, and with a range of side-wall clearances of D to zero. A minimum critical submergence can be expected with side-wall clearances of any reasonable value and with end-wall clearances equal to zero. Efficiency tests with this arrangement may show negligible differences as compared to normal efficiencies, since the inflow can reach the suction from three sides and may not influence the flow pattern in the suction bell to any noticeable degree.

ACKNOWLEDGMENT

Equipment and support for these studies were obtained from Peerless Pump Division of Food Machinery Corporation.

Discussion

A. J. Stepanoff.[6] The unprecedented and unexpected expansion of power plants in the United States since the war's end presented some hydraulic problems in connection with installation of vertical condenser circulating pumps of the propeller type requiring a great deal of discretion on the part of plant designers and pump manufacturers. The basic cause of the difficulty was in the fact that considerably greater capacities were wanted than were provided for in existing intake tunnels or pump cells. A number of new plants having high tunnel velocities and sharp turns to the individual cells experienced similar trouble—pump hydraulic noise and mechanical vibration. The problem is now well understood and several corrective measures have been applied to existing installations with good results.

Model-testing of complete installations, including the inlet tunnel, have been used successfully in diagnosing the faulty installations, as well as proposed plans where trouble had been anticipated. There is a general agreement that rough running of pumps is caused by vortexes at the pump intake induced by a sharp turn in the channel of approach to the pump. Therefore a model does not have to reproduce the pump itself but merely an intake pipe through which water is withdrawn. The velocity distribution can be observed through the lucite channel bottom (and top if the tunnel is under pressure) with the aid of a mirror, and addition of sawdust to water.

From consideration of Froude's number or the expression for the centrifugal forces (which upset the velocity distribution) to maintain a similarity of the velocity distribution, velocity in the model should be the square root of the model-reduction factor. However, since the sense of forces does not change with the rate of flow, it is advantageous to run tests at exaggerated velocities and thus intensify the observable characteristics of the flow.

There are two methods of correcting the field troubles both proved by model-testing and on actual plant installations. In one the flow is stabilized in the impeller approach by giving the channel the shape either of a reducing elbow[7] or that of a vaned elbow with a number of short guides to steady and turn the flow, immediately ahead of the pump inlet.[8]

In the other type of installation attempts are made to improve the velocity distribution at the turn into the cell by proper baffling. Fig. 8(A) of this discussion shows type of velocity distribution observed in a cell as a result of a 90-deg turn; Fig. 8(B) shows three baffles which stabilized the flow sufficiently to eliminate all the noise or vibration. Another pump in the same plant having a longer channel of approach was operating satisfactorily without any baffling. In Fig. 9 is shown an arrangement where water makes two 90-deg turns in succession and the type of baffling required to eliminate the vortexes (baffle *d* was not installed).

In this particular installation the tunnel is under pressure; therefore the flow pattern is essentially two-dimensional. This intensifies the vortexes but makes the baffling more effective. In open tunnels the flow is three-dimensional. While the intensity of the visual characteristics is reduced it requires three-dimensional baffling to steady the flow. Fig. 10 shows an example of the open tunnel and cells.

In addition to a number of short baffles in direction of the flow ahead of each of four pumps, there is a horizontal baffle and a vertical baffle normal to the direction of flow. The latter two

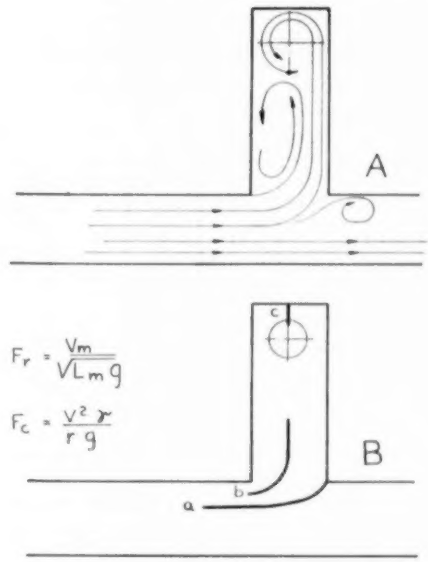


Fig. 8 Suction Sump With 90-Deg Turn
(*a*, *b*, *c*—baffles.)

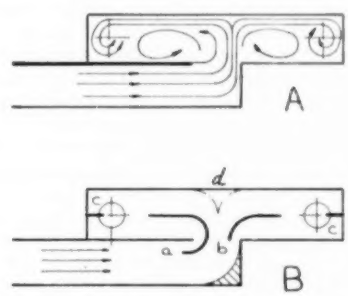


Fig. 9 Suction Sump With 180-Deg Turn
(*a*, *b*, *c*—baffles. Tunnel under pressure.)

baffles cut the high-velocity curls in a vertical plane reflected by the wall normal to the flow from the screen compartments.

In every case it was found beneficial to have the distance between the pump-inlet suction bell to the floor not more than one half of the bell diameter. The distance between the suction-bell edge and the back wall should be one quarter of the bell diameter. A baffle under the bell in direction of the flow is recommended (baffle *c* in Figs. 8 and 9).

The guiding principle in arranging the baffles, as exemplified in Figs. 8 and 9 herewith, became apparent from the model experimentation: each baffle "scoops" its share of the flow from the intake tunnel. This is equally divided by another baffle. When two streams meet at the pump inlet there is no tendency for a vortex formation. In a three-dimensional flow the pattern is more complicated.

Of the two methods of stabilizing the flow in the pump approach channel the second is structurally simpler and does not incur any measurable hydraulic loss chargeable to the pump.

Regarding the vaned elbow as a remedy of the intake-tunnel troubles it may be pointed out that the well-constructed vaned elbow is able to turn the flow 90 deg with a minimum distortion of the velocity distribution, but it will preserve the velocity distri-

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[6] Development Engineer, Ingersoll-Rand Company, Phillipsburg, N. J. Mem. ASME.

[7] "Intake Tunnel Design," by A. I. Ponomaroff, ASME paper No. 50—A-138.

[8] "Suction Intake Design for Vertical Circulating Pumps," by W. W. Weltmer, Midwest Power Conference Proceedings of the Illinois Institute of Technology, Chicago, Ill., 1949, pp. 147-151.

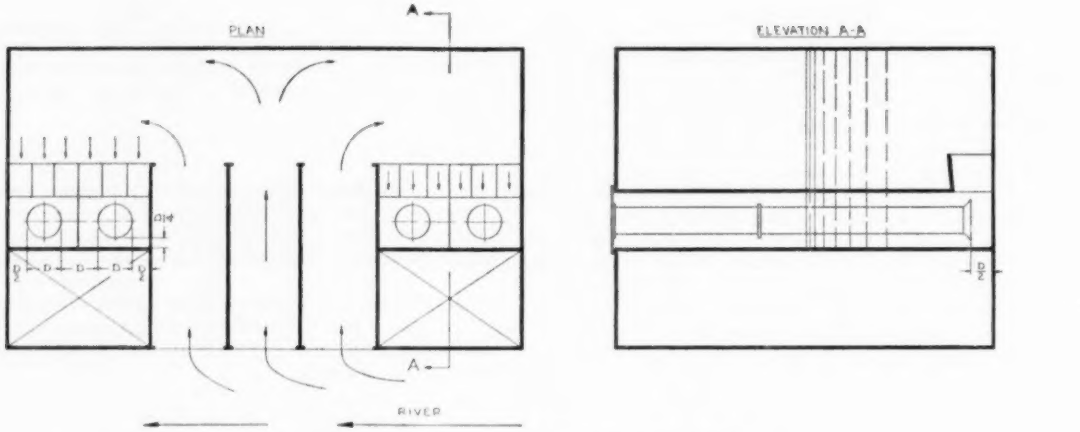

Fig. 10 Suction Sump With 180-Deg-Turn Open Pit

bution disturbed by the shape of the channel of approach to the elbow. Cases are known from a recent experience of vertical pumps developing objectionable vibration with a vaned elbow at the pump inlet.

The present papers[9] of Fraser and Iversen were inspired by the problems faced by the industry since the end of the war as pointed out in the foregoing. Their method of attack is more academic in nature dealing with some fundamental components of the problem without a direct connection with any actual installations, or "trouble" jobs. In many respects both papers are instructive and give some food for thought to the plant designers. Several specific remarks are offered as follows:

The major portion of Fraser's paper[9] is devoted to hydraulic losses in the intake tunnel and the pump immediate approach. The results are only of academic interest to the pump manufacturer, and perhaps even to the operator of the plant. According to accepted testing procedure these tunnel losses are not charged to the pump, and it is not often that pump engineers are involved in the intake-tunnel design. In all cases known to the writer of actual field troubles and discussion of new proposed plants, the question of the pump performance, or tunnel losses was not a consideration—a satisfactory mechanical operation was the only concern. While the beneficial effect of the reducing elbow advocated by Fraser for the shape of the channel of approach (in connection with the scoop) are well established, whether such approach would result in a vibration-free operation of the large units in the field requires experimental proofs, which Fraser does not furnish.

Fraser's recommended proportions and wall clearances for a normal design of a pump-intake sump seem entirely too liberal. Recent model-testing has proved that smaller wall clearances lead to a better flow distribution around the suction bell consistent with the optimum hydraulic performance. In this respect Iversen's tests and earlier tests by Folsom are most illuminating. It turned out that side-wall and bottom spacing of one half of the suction-bell diameter and the spacing of one quarter of the same diameter from the end wall are the optimum setting for the best performance, most stable velocity distribution with a minimum submergence. Perhaps the difference between these figures and those given by Fraser is only apparent—and is a result of different suction-bell sizes in respect to the sizes of the pumping elements. The writer's figures are based on the bell-diameter size approximately twice that of the impeller suction-eye diameters. The spacing from the bottom of one half of the suction diameter should be considered as a maximum. This may be reduced to a minimum of one-quarter bell diameter, if necessary, to equalize the velocity distribution around the bell.

The harmful effect of vortexes may be observed even in the tunnels under pressure where no air is drawn into the suction bell. The fact that such vortexes are not symmetrical with respect to the pump shaft, and do not stay in one place, imposes variable loading on the impeller blades and results in mechanical vibration of the unit.

A minimum submergence to prevent air-drawing as shown by Iversen's tests falls below the self-priming level (top of the impeller hub) and rarely will be encountered in the power-plant application, although there are numerous services where the suction level may be approaching the unsafe minimum. Use of wooden floats to prevent formation of the surface vortexes is an old and effective measure in such cases.

[9] Reference is made to the paper by H. W. Iversen which precedes this discussion, and the paper "Hydraulic Problems Encountered in Intake Structures of Vertical Wet-Pit Pumps and Methods Leading to Their Solution," by W. H. Fraser, which is published in this issue of the Transactions, pp. 643–652.

R. Rankin.[10] As is indicated by the author, information on the literature on this subject is scarce. We who are involved in the practical application of vertical pumps especially appreciate the addition of the practical data which is well presented in the subject discussion. Thousands of dollars are wasted each year because most vertical-pump sump designs are laid out by rule-of-thumb methods which frequently have little basis on actual facts.

The title of the paper indicates that the data is largely applicable to high-specific-speed pumps. However, the bulk of the data is the result of flow through a vertical bell mouth in which there was no pump whatever. Thus the model-test results apply equally well to any specific speed pump having a bell-mouth suction manifold and equivalent flow velocities.

Most engineers laying out a sump for vertical pumps are of the opinion that the greater the clearances of the suction bell from the bottom and sides of the sump the better the performance of the pump will be. They have normally considered the cost of the sump as being the only reason to limit its size. Even though existing literature as cited in the subject paper has indicated advantages from an efficiency point of view in limiting sump clearances, most designers are reticent to design for these optimum clearances. Although there are many unanswered ques-

[10] Chief Engineer, Peerless Pump Division, Food Machinery and Chemical Corporation, Los Angeles, Calif. Mem. ASME.

tions in the subject investigation, both the model and prototype tests indicate a very definite and clear-cut reduction in vortex formation with restricted clearances between the suction bell and the end wall as well as the bottom of the suction and the bottom of the sump.

It is becoming increasingly evident that more economical sump construction may coincide with more efficient pump operation.

The most important question which is not fully answered appears to be a firm knowledge of the relationship between the model and the prototype. In order to get the full confidence of people who are actually going to be making sump designs, it is going to be necessary to get the best possible data indicating the relationship between model tests and prototype tests.

The author states that discrepancies between the model and prototype tests must be found in either surface tensional or frictional force influences. However, based on the information presented in the paper, the writer does not see how we can completely rule out the possible effect of the rotating impeller in the pump which was present in the prototype test and not in the model. Tests with a pump having a suction bell should be compared to a pipe having an identical suction bell but not including the pump and should be checked under identical flow conditions to verify the theory that vortex action in the sump is completely independent of pump rotation. In such a direct comparison it will also be important to investigate the possibility of operating the pump away from the designed condition where recirculation may affect vortex action to a much greater extent than at the design point of the pump. It is often not at design point but at high-capacity flows that vortex action is particularly troublesome. Whether this is due purely to higher rates of flow in the sump or has some relation to the pump and suction manifold design should be established.

We have conducted several model tests using plastic models and a suction bell attached to a pump remote from the suction bell. Although the basic results have been as expected, we have discovered many things in the model tests which have led us to realize that it is practically impossible to analyze theoretically a sump design which deviates appreciably from configurations with which we have had experience. By using a transparent plastic model we have been able to observe the flow much more clearly than is evident from surface indications. We are encouraging engineers to make model tests on all multipump sump designs which do not conform to normal configurations.

One of the worst sump designs which we have investigated has been that of a large horizontal completely enclosed circular conduit feeding a circular caisson in which a single pump is installed. Because of the ease of construction, this is not an uncommon sump configuration but is a particularly undesirable one.

No mention is made by the author of the possibility of vortex formations affecting the cavitation characteristics of the pump. Although these are completely different phenomena, it is quite possible that where a pump is operating in a borderline region in relation to cavitation the pulsation and disturbance created by the vortex formations may cause cavitation and be particularly detrimental to the pump. It is realized that the separate investigation of either of these factors is far from complete, but the combined effect of both should not be overlooked on ultimate future studies.

Author's Closure

Dr. Stepanoff presents some useful information on various arrangements of guide vanes to stabilize inflow from intake tunnels into pump suction bells. His suggestions relative to baffling in open pumps, as shown in Fig. 10, with the horizontal and vertical baffling, are interesting applications of flow stabilizers.

The use of horizontal baffles to eliminate local vortexes in sumps needs further investigation. Some recent work by the author, in sumps of the arrangement of Fig. 1, showed that a horizontal baffle between the end wall and the pump column prevented local vortex formation downstream of the pump column. The baffle spanned the sump at the minimum desired pumping level, and was cut out around the pump column to permit free vertical removal of the pump.

Dr. Stepanoff states that the pump designer is not concerned with the initial intake tunnel or channel design as it influences the system performance of head and efficiency. More attention should be given to the over-all system performance, both by the sump designer and the pump designer, to provide for good inflow to the pump without unwarranted hydraulic losses or installed "trouble shooting."

Some sump arrangements must contain tortuous channeling because of the system space restrictions. As Dr. Stepanoff points out, in many cases simple model tests will insure proper sump design.

Mr. Rankin's comments are appropriate. Further investigation of the problem of sump design is necessary to determine the possible effects of prerotation induced by the pump inflow, particularly at flows in excess of the design flow, on the formation of vortexes.

Vortex formations in the sump, which are not strong enough to initiate an air core, will influence the cavitation characteristics of the pump. The velocities in the rotating vortex are larger than normal inflow velocities. At the entrance to the pump impeller the vortex velocities result in local low-pressure regions which promote cavitation. Thus, the necessary submergence to prevent cavitation will be greater than that determined from normal inflow tests. Sump design to eliminate the vortexes also reduces harmful cavitation influences.

Hydraulic Problems Encountered in Intake Structures of Vertical Wet-Pit Pumps and Methods Leading to Their Solution

By W. H. FRASER,[1] HARRISON, N. J.

The design of intake channels and tunnels for the efficient operation of wet-pit pumps is considered. Basically the problem is that of determining the flow pattern and losses in the suction intake and predicting their combined effect on the pump performance. Because of long distances, obstructions, and diversions, faulty flow often occurs. The resulting difficulties are explained and the methods employed in solving the attendant problems are outlined.

THE flow of water in open and closed channels has demanded the attention of engineers from the earliest investigators in hydraulics down to the present day. Antoine de Chezy, in 1775, determined experimentally that turbulent flow in an open channel can be described by the relation $V = C\sqrt{RS}$, where V is the average velocity of the flow in the channel, S is the slope of the channel, R is the hydraulic radius, and C is an experimental coefficient. Wilhelm Kutter, in 1869, proposed a roughness index n, now called Kutter's n, in conjunction with an empirical equation for determining a value of C. In 1890, however, Robert Manning published a much more usable relation for the evaluation of C, also containing Kutter's n. Chezy's basic relation was later found to apply also to flow in a closed conduit and Henri Darcy and others have shown that the head loss in a closed conduit is proportional to

$$fL/D\, V^2/2g$$

where L is the length of the conduit, D the diameter, $V^2/2g$ the velocity head, and f, a friction coefficient. The coefficient f has been found from exhaustive tests to be a function of the Reynolds number and the roughness. These flow problems have assumed a new importance to the pumping engineer within the past 15 years in the form of intake channels and tunnels for vertical wet-pit pumps. The vertical wet-pit pump is assuming greater importance each year mainly because of its economical use of space and the need for greater submergences as units of higher specific speed are being designed.

The problem is basically that of determining the flow pattern and losses in the suction intake and predicting their combined effect on the pump performance. Unfortunately, this is seldom the case as the water must often flow relatively long distances from the source and, because of building construction, undergo a restriction or change in direction before it is available at the pump suction. The problems resulting from faulty flow or the inability to predict flow and the methods leading to their solution will be discussed in this paper and it is felt that too much emphasis cannot be placed upon the importance of the intake design as it reflects itself in pump performance. Any wet-pit pump is no better than its intake performance and as much study and investigation must go into its design as into the pump itself.

Action of Typical Pump Installation

No particular problem exists when only friction losses of the intake need be considered, as such losses can be predicted with reasonable accuracy. A typical installation with a good intake

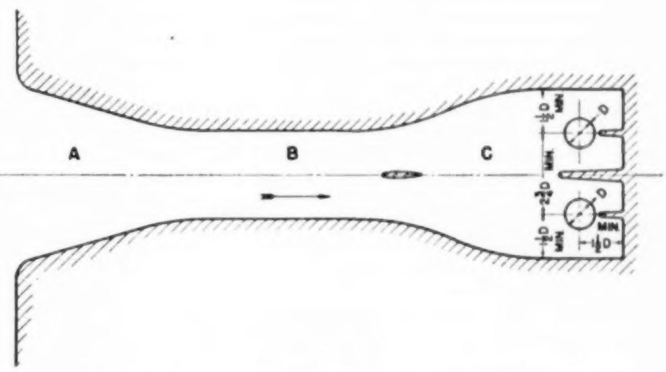

Fig. 1 Typical Pump Installation and Intake Design

design is shown in Fig. 1. Two pumps are required at some distance from the source and the combined capacity of the two pumps flows through a single tunnel B and is then discharged into a forebay C. The entrance A to the intake should be designed as a gradual contraction and losses as small as 0.1 of the tunnel velocity head can be realized. With recommended tunnel velocities of 3 to 4 fps, the entrance losses and the tunnel losses are very small provided the walls are reasonably smooth and the length not excessive. In this connection the rate of marine growth on the tunnel walls should be fairly well known so that preparations can be made to remove such growths when the increased losses and increased velocities, due to a decreased tunnel cross section, affect the pump performance. With a well-designed enlargement to the forebay C the losses can be held to 0.3 of the change in velocity head from the tunnel to the forebay, but there is the danger that the flow will cling to one side of the forebay and feed one pump at the expense of the other. This condition can be improved by installing a baffle or splitter at the enlargement to distribute the flow equally to each pump. With an efficient enlargement there will be a rise in the static level in the forebay equal to the change in the velocity head from the tunnel to the

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forebay less the losses. This rise is desirable as the static head against which the pump must operate is reduced by the amount of this rise with a proportional reduction in the power input at the pump shaft. It is also possible, with formed pump suctions, to utilize a portion of the velocity head of the channel flow as pumping head without recovery to static head at the pump suction. The use of formed pump suctions will be discussed further in the latter portion of this paper on high tunnel velocities.

If, as in many installations, the water level at the source varies seasonally, the tunnel will run full at high levels, but may flow with a free surface in the tunnel at low levels. These two conditions must be studied separately as the determination of the flow in a closed conduit applies at high levels, but that of open-channel flow applies at low water levels.

In Fig. 1 the recommended practical arrangement is shown as a function of the suction-bell diameter D. Figs. 2, 3, and 4 present the recommended practical arrangements for three typical installations. Fig. 4, however, may not operate satisfactorily with high-specific-speed pumps owing to the flow disturbances of the upstream pumps. Furthermore, the high-specific-speed pumps are particularly sensitive to the suction conditions and, to assure the expected performance, baffles should be installed between the suction-bell rim and the back wall in Figs. 2 and 3. Particular reference should be made to the profile of the approach channel as shown in Fig. 2. When the bed of the channel slopes toward the pump, the flow is accelerated and the free surface is depressed as a result of the increased velocities. The pump submergence is reduced by this depression and the approach velocities may exceed the suction-bell velocities and increase the entrance losses.

In installations of the type shown in Figs. 1, 2, 3, and 4, a portion of the incoming flow continues until it strikes the back wall. Owing to this impact, the level builds up and the flow is reversed back toward the pump. The level immediately downstream from the pump is lowered by the wake caused by the diversion of flow around the pump body in addition to the drawdown caused by the reduction in pressure at the pump suction. The difference in levels between that at the back wall and the wake of the pump

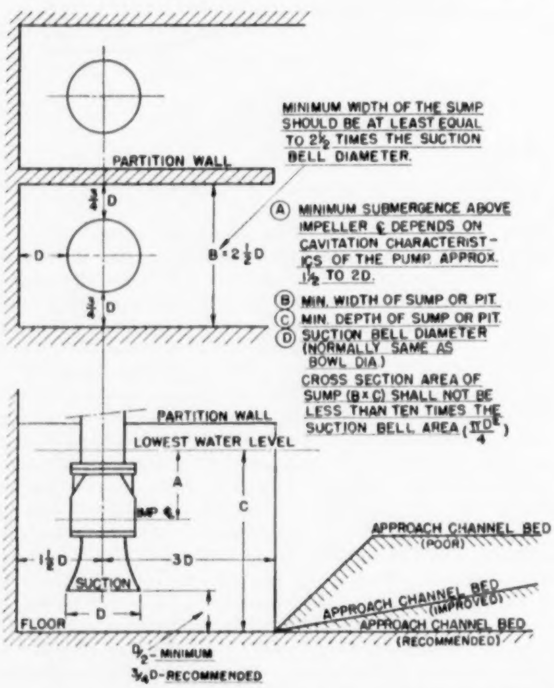

FIG. 2 CHANNEL AND PIT DESIGN FOR VERTICAL TURBINE PUMPS

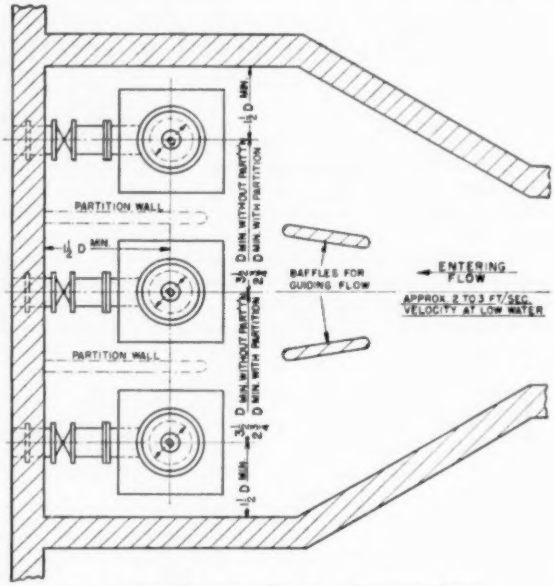

FIG. 3 SUMP PLAN FOR VERTICAL TURBINE PUMPS
(Recommended arrangement when suction pits are restricted.)

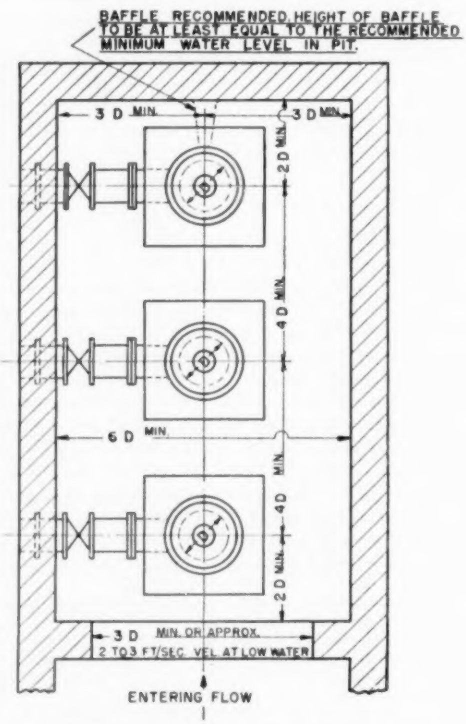

FIG. 4 SUMP PLAN FOR VERTICAL TURBINE PUMPS
(Not recommended when suction pits are restricted.)

may create severe turbulence which serves to destroy any vortexes that might develop. If the pump is in an extended channel, however, with additional pumps downstream to increase the flow sufficiently, these vortexes can develop and result in unsatisfactory pump performance.

Theodore von Kármán has demonstrated that a definite arrangement of vortexes and eddies is created behind a stationary cylinder in straight channel flow. It has been demonstrated further that the flow separates from the cylinder with a definite frequency, and if this separation frequency approaches the natural frequency of the cylinder, severe vibrations may result. This condition may exist in suction channels, as the pump body is essentially a cylinder, and to prevent such induced vibration of the pump with the possibility of drawing air into the suction, such installations should be avoided. In other words, each pump should be fed directly from the source, and not from a flow that has been disturbed by another pump upstream.

As the flow between the pump column and the back wall is highly turbulent, any quantitative analysis of these conditions is uncertain. An indication of the magnitude of these disturbances can be approximated, however, by determining the impact of the water against the upstream side of the pump column and against the back wall. The column and the back wall are stagnation points and the force at these points is proportional to the time rate of change of momentum of that portion of the flow affected. The mass of water affected in one sec is

$$m = \frac{w}{g} AV$$

where A is the area in square feet normal to the flow and V is the average velocity in feet per second. Since momentum is equal to mV, the impact force is equal to

$$\frac{w}{g} AV^2$$

A coefficient C_d, comparable to the drag coefficient used in aerodynamic studies, can be applied and averages 0.6 to 0.8 for the pump column and close to 2.0 for the back wall. The Reynolds number encountered in suction intakes always exceeds 10^6 and thus the boundary-layer flow is always turbulent with fairly constant values of C_d. It follows, therefore, that the greater the value of

$$C_d \frac{w}{2g} AV^2$$

the more severe will be the turbulence in the vicinity of the pump and the greater will be the forces on the pump column. In this connection, it may be possible, with further study, to determine accurately the inception of vortex formation in this type of intake structure.

Straight-Flow Suction Channel

In straight-flow suction channels the formation of local vortexes at the free surface and extending into the pump suction has often been observed. In themselves local vortexes are of little consequence to pump performance, but if they develop sufficiently, the core will permit large quantities of air to be drawn into the suction. The adverse effect of air entrainment is not limited to the pump performance alone, for the free air will accelerate the corrosion and erosion of the pump, the equipment it serves, and the piping. Vortexes are generated at adjacent surfaces of water flowing at different velocities and their severity and persistence will depend upon the velocity gradient. It is thus apparent that all velocity changes that occur in the intake structure should be as gradual as possible to prevent the existence of conditions conducive to such vortex formation. Owing to the drawdown in the vicinity of the pump, which in turn affects the frictional slope of the incoming flow, a velocity differential will exist between that portion of the water that is diverted to enter the pump suction and that portion that flows downstream past the pump. With sufficient submergence over the suction bell and a gradual acceleration of the flow into the bell, the velocity gradient will not be sufficient to form or to sustain a vortex. A theoretical analysis of this phenomenon is extremely complicated and uncertain as the core of the vortex is seldom perpendicular to the free surface for an appreciable depth. In fact, the vortex mouth may form a considerable distance from the pump and the core extend at a relatively flat angle into the pump suction. Experience indicates that the formation of vortexes, in a correctly designed channel, is dependent upon the submergence over the bell, the velocity into the bell, and the velocity of flow in the suction channel.

Vortex at Pump Suction

The most serious problem encountered in suction intakes is that of a persistent and large-scale vortex at the pump suction. The design specific speed of a wet-pit pump is dependent upon straight-through flow into the suction bell, and if this pattern is disturbed the capacity and head at maximum efficiency will be affected. If the water at the suction rotates in a direction opposed to that of the pump rotation, the pump output will increase with a proportional increase in power required to produce this condition. Since the pump head is dependent upon the algebraic sum of the angular momentum at the suction and that produced by the impeller, it is apparent that a negative angular momentum of the flow at the suction, as a result of counter rotation produced by the intake structure, will increase the pump output.

A number of cases are on record where the drivers of wet-pit pumps have been overloaded severely which could be attributed solely to this rotation in the suction intake. Conversely, if the rotation of the water is in the same direction as the pump rotation, the pump output will decrease with a reduction in power, and may not satisfy the anticipated conditions. The formation of a large-scale vortex is usually associated with an intake design that causes a change in direction of the flow before it enters the pump suction.

In a multiple-unit installation of identical pumps a number of the pumps may operate satisfactorily, but the remaining units may overpump or underpump in an apparently haphazard fashion. Upon investigation, however, it will be evident that because of the location of the various units the suction conditions are not duplicated and overpumping or underpumping occurs depending upon the magnitude and direction of the swirls. It is thus apparent that identical pumps cannot be considered as duplicates unless the suction-flow conditions to each are also duplicated.

It has been learned from field experience and through model studies, that if the change in direction of the water is not too severe, a baffle placed between the suction-bell rim and the back wall in line with the incoming flow, as shown in Fig. 1, will assure satisfactory operation. The baffle should be placed as close to the suction bell as possible and extend to the surface of the water in an open channel or to the roof of the tunnel in a closed system.

Experience has shown that the effects of the large-scale vortex, if not too severe, can be controlled with baffling. Larger and more complex installations involving a number of pumps generally operate at higher tunnel velocities and the simple application of baffles may be ineffective.

Fig. 5 shows a typical installation of this type in which the pumps are placed in individual wells out of the main stream flow. To illustrate, if each of the six pumps shown has a design capacity

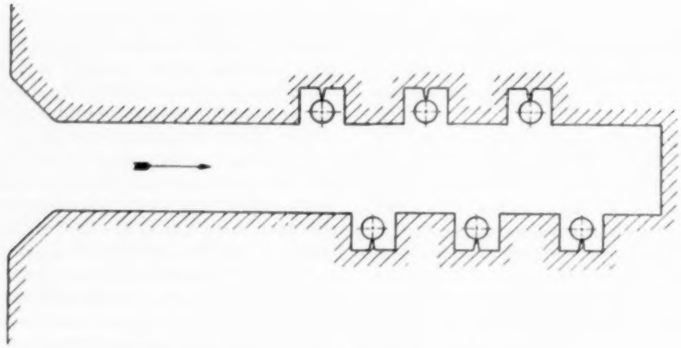

Fig. 5 Illustrating Suction-Scoop Study

of 25,000 gpm, the tunnel flow at the first well is 150,000 gpm at a tunnel velocity of 6 fps. The velocity head represented by this velocity tends to maintain straight flow through the tunnel and the flow into the wells will be proportional to the difference in the pressure in the tunnel and the level in the well. The level in the well is determined by the drawdown of the pump and will increase until a sufficient differential exists to divert the required capacity into the well. The reduction in level, however, will manifest itself to the detriment of the pump in at least three forms:

(a) The suction head available at the impeller is reduced, and if less than that required by the pump, cavitation will occur. This condition cannot be tolerated for any length of time as it will affect the pump performance adversely and accelerate the destruction of the leading edge of the impeller vanes.

(b) That portion of the flow which is diverted into the well still retains a component of its forward velocity and produces a severe swirl that cannot be controlled effectively by baffling.

(c) The reduction in level will increase the total pumping head by increasing the static head between the suction and discharge levels. This is an example of uncontrolled flow at high velocities and can be improved only by providing a means to utilize a portion of the energy of the tunnel flow and guiding the flow evenly to the impeller. The usual practice is to provide a scoop or contracting elbow located in such a manner that as much flow is diverted as required by each pump and yet does not restrict the flow to the downstream units.

Formed suctions have proved to be very effective with high-velocity flows and, when it is realized that a flow of 150,000 gpm at a velocity of 6 fps represents 21 hp, it is apparent that every effort should be made to utilize this power with a minimum of loss. The formed intake structure, however, will increase the cost of the installation materially and the engineer must decide whether or not the sacrifice in pump performance warrants the additional construction costs.

Model Tests of Intake Structure

The most effective method for the study of these problems is by model tests of the intake structure where controlled conditions can be maintained and alterations made at little cost. Model studies, however, are not infallible, and considerable skill and judgment must be exercised in their design, operation, and interpretation of results. Such models have been designed, built, and tested, and the results when applied to the prototype have proved effective. A model of the complete intake structure, from the inlet to the pump suction, is seldom necessary and the usual practice is to model that portion where the most severe conditions occur and to select as large a scale as is practicable.

Models of intake structures fall into two general classifications, models of open-channel intakes and models of closed conduits or tunnel intakes. The surface conditions in an open channel follow Froude's law which states that the surface disturbances can be described by Froude's number which is equal to the average velocity V at a representative cross section divided by the square root of a representative linear dimension L and the gravitation constant g. This can be expressed mathematically as

$$F_r = \frac{V}{\sqrt{Lg}}$$

It is further recognized that to produce comparable conditions in two geometrically similar structures of different size, Froude's number must be held constant. Now if L_m is a linear dimension of the model and L is the corresponding linear dimension of the prototype, the scale factor is L_m/L. Further, the Froude number of the model is

$$F_r = \frac{V_m}{\sqrt{L_m g}}$$

and of the prototype is

$$F_r = \frac{V}{\sqrt{Lg}}$$

and it follows that with constant Froude number

$$V_m = V\sqrt{\frac{L_m}{L}}$$

Modeling of the pump suction to maintain geometric similarity requires that the suction bells and the flow pattern in the model and the prototype be similar. The ratio of the model and the prototype velocities, however, need not be related to the scale factor to maintain geometric similarity.

It would appear that a model designed for constant Froude number, i.e.

$$V_m = V\sqrt{\frac{L_m}{L}}$$

will satisfy the model relations for both the surface flow conditions and the pump suction. This assumption is reasonable if the model scale is not too small and the prototype velocities sufficiently high.

As the model scale decreases, the model flow velocities become very low as compared to the prototype and the results are unreliable. Satisfactory results have been obtained, however, if the

model is designed with the same flow velocities as in the prototype. With velocities higher than required for a constant Froude number the eddies and turbulence in the model will be more severe than in the prototype and it is reasonable to assume that if these adverse flow conditions can be corrected in the model, the same measures will be effective when applied to the prototype.

Flow in Closed Conduit

The flow in a closed conduit or tunnel can be described in a similar manner by use of the Reynolds number which is equal to the product of a representative linear dimension and the average velocity over a cross section normal to the flow divided by the kinematic viscosity of the fluid being considered. Now, if the inertia forces predominate and the viscous forces are not sufficient to invalidate the results, the Reynolds number will vary directly as the model scale and as the fluid velocity. The modeling of hydraulic machinery has demonstrated that in most instances the viscous forces are negligible and the so-called Reynolds-number effect can be neglected. It is reasonable to assume that the same procedure can be adhered to in the modeling of suction tunnels and that, unlike the open-channel models, the suction-tunnel and pump flow conditions follow the same dynamic similarity laws. Thus all velocity patterns in the geometrically similar model will be essentially identical to those in the prototype even at widely different Reynolds numbers. The practicability of this procedure has been borne out where effective changes have been made in intake designs based upon information obtained from studies of such approximate models.

Suction Scoops

To study the effectiveness of suction scoops in an installation, as shown in Fig. 5, with varying tunnel velocities, a $^1/_{16}$-scale model was built with the same velocities as in the prototype. Refer to Fig. 6 for a diagrammatic sketch of the model. To attain the desired velocities past the first well, a true model would have included additional pumps, but modeling of the first two wells only was considered sufficient to obtain the essential information. The model consisted of a crib which served as a reservoir to maintain a constant static head on the tunnel comparable to the actual river level. The No. 1 well was placed a sufficient distance from the junction of the tunnel and the crib so that the inlet conditions into the tunnel would not affect the readings at the first well. The desired tunnel velocities were obtained by an auxiliary pump which took its suction from the end of the tunnel and recirculated the water back to the crib. By throttling the discharge of this pump it was thus possible to vary the tunnel velocities over a wide range. It is very convenient in this type of model to use siphons with modeled inlets to duplicate the pumps.

Fig. 6 shows the modeled scoop in place in the No. 1 well and the orifice meter in the down leg of the siphon to measure the flow rates. The siphon head, measured by means of a Bourdon gage as shown, is that head required to produce the flow rate through the suction bell and siphon system. The flow removed by the siphons was replaced by make-up water in the crib to maintain a constant level throughout the tests. Table 1 gives the pertinent specifications of the prototype and the corresponding model values.

To obtain a comparison of the relative merits of the suction bell and the scoop suction, the change in capacity and siphon head with each suction design at a constant valve setting of the siphon was obtained. It is apparent that the greater the turbulence and losses into the well, the lower will be the capacity of the siphon and the greater will be the required siphon head. It follows that all losses in the siphons themselves must be isolated and this was done by plotting the static levels in the wells against

TABLE 1 PROTOTYPE AND MODEL DATA

	Prototype	Model
Tunnel cross section	8 ft × 15 ft	6 in. × $11^1/_4$ in.
Well opening	8 ft × $8^1/_2$ ft	6 in. × $6^3/_8$ in.
Well size	$9^1/_2$ ft × $8^1/_2$ ft	$7^1/_8$ in. × $6^3/_8$ in.
Pump capacity—each	34500 gpm	135 gpm
Suction-bell diameter	44 in.	$2^3/_4$ in.
Scoop inlet	2 ft × 4 ft	$1^1/_2$ in. × 3 in.
Static head on tunnel	15 in.	$3^3/_4$ ft

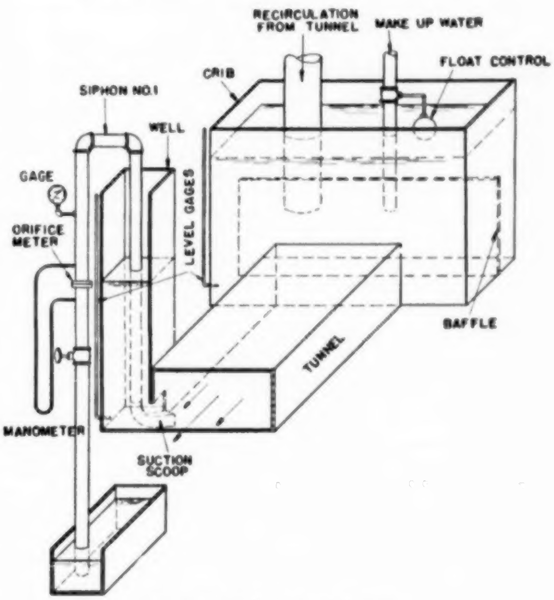

Fig. 6 Model Suction Tunnel

the siphon flows with tunnel velocities equal only to those caused by the siphon flow. This plots as shown in Fig. 7 with the suction-bell inlet, and in Fig. 8 with the suction-scoop inlet. Using these curves as a calibration for each, any deviation in capacity at constant siphon heads will indicate the effectiveness of the suction design.

Examination of Fig. 7 with the bell suction shows a marked decrease in capacity for pumps Nos. 1 and 2 up to about $3^1/_2$ fps tunnel velocity, and then with a further increase in tunnel velocity, the curves approximately parallel the calibration curve up to velocities of 9 to 10 fps when the deviation begins to increase. Throughout the range of velocities tested, with the exception of the low tunnel velocities, there is little difference in performance between the Nos. 1 and 2 pumps.

Fig. 9 shows the loss in capacity plotted on a percentage basis against tunnel velocity. The single curve shown is an average of the loss in capacity of the Nos. 1 and 2 pumps. It must be remembered in the application of these curves to the prototype that the percentage loss in capacity reflects losses into the well only, and gives no indication of the magnitude or direction of the swirl in the well and its effect upon the pump performance.

Visual examination during these tests revealed severe swirling in both wells even though a baffle had been installed between the suction bell and the back wall of the well. Readings of the drawdown in each well were taken and the feet drawdown is plotted against tunnel velocity in Fig. 10. The curve applies for both the Nos. 1 and 2 wells as very little difference was noted between the two. The velocity head in the tunnel also is plotted on the same scale and the difference between the velocity head and the drawdown represents the head loss incurred with a 90-deg turn of the water into the well. It can be seen from

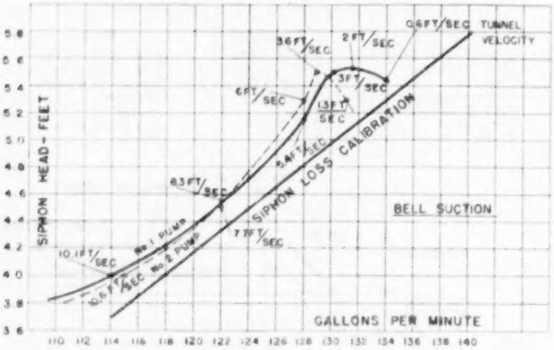

Fig. 7 Siphon Loss With Bell Suction

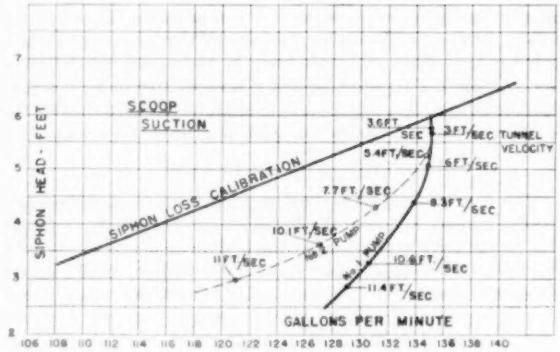

Fig. 8 Siphon Loss With Scoop Suction

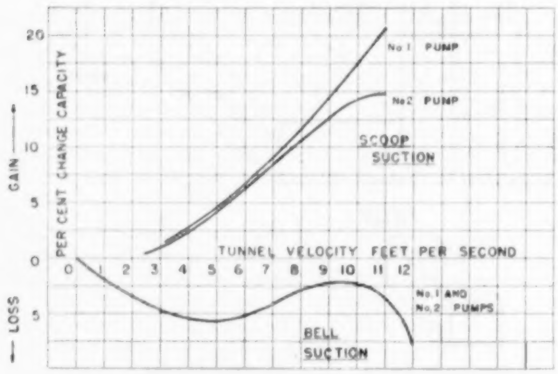

Fig. 9 Comparison of Losses With Scoop Suction and Bell Suction

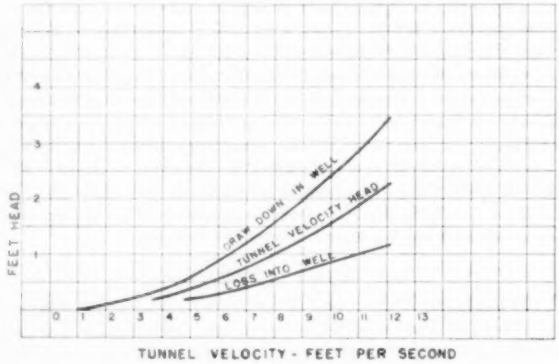

Fig. 10 Drawdown and Head-Loss Curves

this curve that a drawdown of $1^1/_2$ ft at a tunnel velocity of 7.8 fps, which would be of the same order of magnitude in the prototype, would be quite serious with a low-head pump as it would increase the pumping head and decrease the available submergence by the same amount.

In contrast to these curves is that in Fig. 8 where the same test was run with the suction scoop in place. It will be noted that there is a gain in capacity as the tunnel velocity is increased with an appreciable spread between the Nos. 1 and 2 pumps.

Fig. 9 shows this increase as a percentage rise in capacity plotted against tunnel velocity. It is apparent from these curves that much is to be gained by the use of the suction scoop which utilizes a portion of the impact velocity of the tunnel flow over the suction-bell design, and, with performance data of this nature, the problem then resolves itself into a cost study of the increase in tunnel construction to reduce velocities, if the suction bell is to be used, as against the cost of the scoop construction which will operate satisfactorily with the high tunnel velocities.

Conclusion

As mentioned previously, recent growth in the use of intake structures for wet-pit pumps has necessitated more careful study, and although a great deal of experience and understanding of the problems has been gained over the years, new problems are arising continually which will demand the close attention and co-operation of the pump manufacturers and of the plant design engineers to arrive at satisfactory solutions.

Acknowledgments

The writer is indebted to the Worthington Corporation for the release of the data of the model tests presented in this paper.

Discussion

D. W. R. Morgan, Jr.[2] The author is to be congratulated on his excellent and most timely presentation of a subject which is of great interest both to pump manufacturers and users.

The problem of intake structures for vertical wet-pit pumps is relatively new and has been accentuated recently by the use of high-specific-speed pumps designed for relatively high velocities which require uniform flow in the suction tunnel for satisfactory operation.

Our experience with condenser circulating pumps indicates that a reduction of circulating-pump capacity often can be traced to unsatisfactory flow in the suction tunnel. Many times this results in severe pump vibration, bearing failures, and excessive wear at shaft journals. For many years we have found that the use of small scale models of pumps and suction tunnels in the laboratory permits observation of flow conditions at the pump inlet and evaluation of the effects of abnormal flow conditions on pump performance. Baffling, water deflectors, and other corrective measures developed with the aid of models have been applied successfully in many cases. However, the use of a siphon, as suggested by the author, often does not show the true flow

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conditions existing in the tunnel. In a high-specific-speed pump the runner produces a disturbance which cannot be duplicated through the use of a siphon in model-testing. For this reason it is our practice in tunnel studies to use model pumps of approximately the same specific speed as the prototype.

The rapid developments in the steam-turbine art have permitted the installation in existing plants of turbine-generator units of much larger capacities than originally contemplated. The tunnel problems associated with such installations in many cases have been quite complex as a result of increased water velocities and the use of high-speed vertical pumps necessitated by space restrictions.

We are in complete agreement with the author that the design of intake tunnels is so closely related to pump performance that the co-operation between the pump manufacturer and those responsible for intake-tunnel design is essential.

D. R. Rankin.[3] It is helpful to have added to our present information the data on sump designs and the discussion included in the paper. Carefully following the recommendations presented by the paper undoubtedly would improve greatly the average sump design used for vertical wet-pit pumps.

There are several details in relation to the over-all sump designs as indicated in Figs. 1, 2, 3, and 4 of the paper, which differ from the results which we have obtained in our own model and field tests and our interpretation of existing literature. The minimum clearances indicated in the author's paper do not appear to us to be substantiated by existing literature (Folsom,[4] Richardson,[5] Stepanoff,[6] Kerr and Moyer,[7] and Iversen[8]).

Since no model or field-test data are presented in the paper to show in a quantitative way the relative values of the sump configurations indicated, it is impossible to evaluate the validity of the minimum clearances shown. The writer does not mean to imply that the proper pump operation cannot be obtained in following the recommendations of the author. However, considerable savings in the construction of the sumps and possible improved performance and decreased vortex formation would result from using smaller clearances. It is particularly for this reason that encouragement should be given to further study and model-testing so that any discrepancies between the recommendations of minimums could be eliminated.

The most valuable new contribution of the author is the latter part of his paper which presents specific model-test information in relation to a multipump installation having a very high-velocity entrance. The data presented indicate definite improvement in using the suction scoops when the high-velocity inlet of the configurations indicated in Fig. 5 of the paper, is used. It is pointed out by the author that even though a baffle has been installed between the suction bell and back wall of the well severe turbulence resulted. It would be of interest to learn how the sump design, as indicated in Fig. 5, requiring velocities of 6 fps in the main channel was developed. If the sump had been designed for velocities in the recommended range of 3 fps the suction scoops would not be necessary. It also should be emphasized that the suction scoops are not of appreciable value where the channel velocities are 3 fps or less, which is the normal recommended velocity range for good sump design. Thus an expedient of the suction scoop normally would find practical application only when good sump practice could not be followed owing to excessive increase in capacity because of unforeseen expansion or similar expediencies.

[3] Chief Engineer, Peerless Pump Division, Food Machinery and Chemical Corporation, Los Angeles, Calif. Mem. ASME.

[4] "HP-14, Suction Conditions," by R. G. Folsom, Pump Testing Laboratory, Technical Memorandum No. VI, University of California, December, 1940.

[5] "Submergence and Spacing of Suction Bells," by C. A. Richardson, Water Works and Sewage, Reference and Data, Part 1, Water Supply, 1941, p. 25.

[6] "Centrifugal and Axial Flow Pumps," by A. J. Stepanoff, John Wiley & Sons, Inc., New York, N. Y., 1948, pp. 363–366.

[7] "Hydraulic Engineering Problems at Southwark Generating Station," by S. L. Kerr and S. Moyer, Trans. ASME, vol. 64, 1942, p. 539.

[8] "Studies of Submergence Requirements of High-Specific-Speed Pumps," by H. W. Iversen, published in this issue of the Transactions, pp. 635–641.

C. R. Wilderson.[9] The author's discussion of the problems encountered in the design of intake passages to sumps for vertical wet-pit pumps has emphasized again the importance of proper flow patterns at the pump-intake bell if good efficiency and operating characteristics are to be achieved. Partition walls and guide baffles will assist greatly in producing the desired improvement in flow pattern, and the use of models in determining water-passage neat outlines will result in improved sump designs. However, the general statements in the discussion do not give the designer a definite procedure or principle upon which to base his design, and, unless he is already skilled in hydraulic water-passage design, the discussion may lead to an elaborate design which will be little better than a simple rectangular structure.

It has been the practice of the Bureau of Reclamation to design intake and water passages by using hydraulic laws manifested by the performance of jets issuing from orifices, flow through Venturis and vortex flow, as design criteria. Model tests are used to confirm the application of fundamental design principles rather than to provide the basic concept through a progressive series of experiments. Should the model tests reveal the presence of flow conditions differing from those predicted for the prototype, it indicates that the basic design principles used were applied incorrectly and a revision of the application to the design is necessary. Once the method of design is established it may be applied to similar problems.

The model tests on the scoop-type intake are interesting and serve to confirm the author's conclusion that the kinetic energy of flow may be utilized to improve the pump performance. The tests give an indication of what might be expected of a properly formed intake structure. It would be interesting to know what effect the shape and size of these intakes have upon pump performance.

The author is to be commended upon his description of model-testing. It provides an excellent procedure that may be followed should one wish to duplicate the model tests or to perform tests of a similar nature.

Many solutions have been offered for the hydraulic problems encountered with vertical wet-pit pump intake structures. The majority of the problems arising are the result of eddies or swirls and the formation of vortex trails. The consensus has been that, to prevent the formation of vortex trails, straight, uninterrupted flow to each pump should be provided, the suction bell should be located at a sufficient distance from the sump walls so that the flow to the pump is not affected materially, and the suction bell should be submerged to such a depth that the formation of vortex trails is minimized. When more than one pump is located in the same sump it is difficult to meet all these conditions. Baffles between pumps and suction nozzles other than the conventional suction bell have been suggested as possible means of preventing vortex formation.

Fig. 11 of this discussion shows a pumping-station design with three vertical pumps located in a wet-pit intake. The pumps are placed in echelon diagonally to the direction of stream flow.

[9] Mechanical Engineer, U. S. Bureau of Reclamation, Denver, Colo. Jun. ASME.

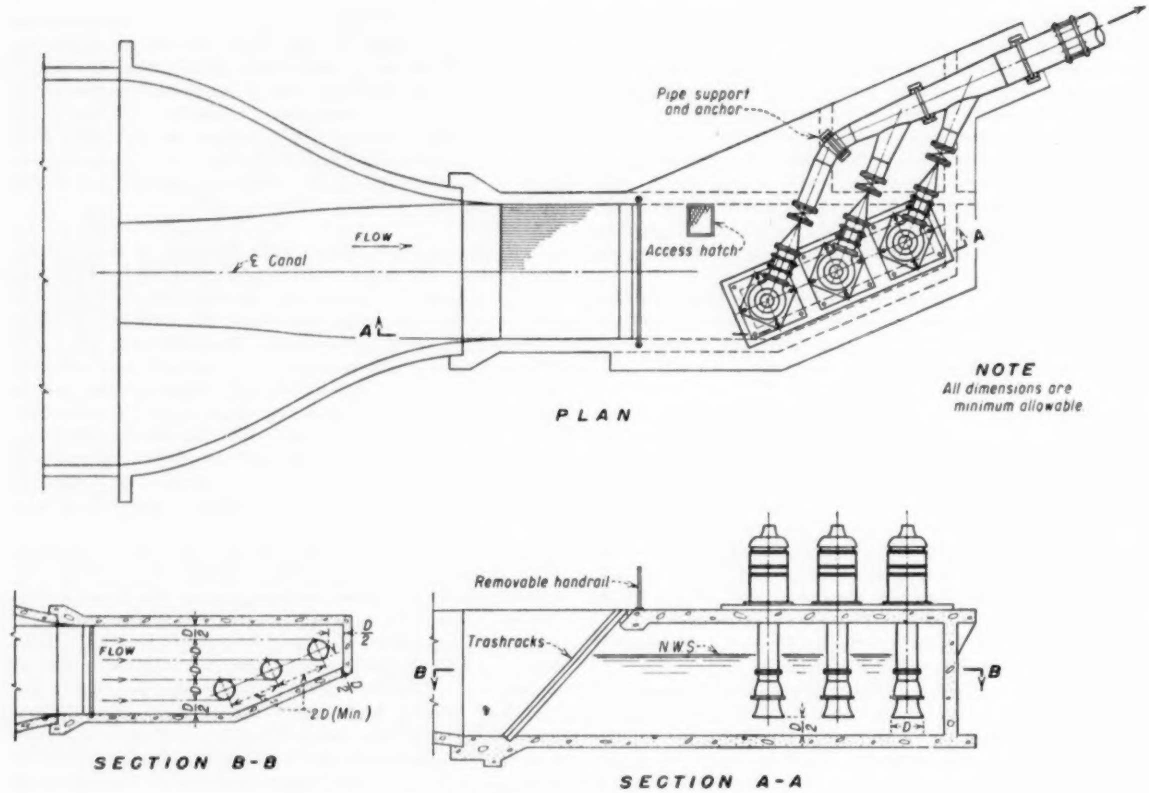

Fig. 11 Multi-Unit Pumping Plant With Echelon Setting

Section B-B illustrates that each suction bell is situated so that there is a straight, unobstructed flow to each, and there is no appreciable change in the velocity of approach. This is an important consideration. The wall and bottom clearance are those that have proved satisfactory and reasonably efficient, both in laboratory practice and field installations as discussed by H. W. Iversen.[8] The test results presented by the author and by Mr. Iversen indicate that the side wall, end wall, and bottom clearances recommended by A. J. Stepanoff for a vertical pump suction bell located in a rectangular sump would not affect the pump efficiency adversely when compared to a pump located in a very large sump, although the test results presented by Mr. Iversen do not appear to be adequate to permit the formation of definite conclusions for optimum clearances or settings. However, it appears probable that an increase of efficiency would be possible by an optimum clearance slightly greater than that recommended by Stepanoff. Greater than optimum wall clearances may be used but constitute added cost in the construction of the sump. The spacing of the pumps is given as a minimum distance and may be increased if necessary for other than hydraulic purposes. A large number of units may be installed in a sump of this design.

The advantages of the design shown in Fig. 11 are as follows: Unobstructed flow to each pump; a narrower, less expensive sump is possible and the sump is of simple shape; difficult concrete forms are not required and baffles or transitions are unnecessary. A possible disadvantage of the proposed design is that the surface drawdown from the upstream pump might tend to starve a pump further downstream. Care should be exercised that the velocity of approach is not too great. Section B-B in Fig. 11 shows that the channel of flow (flow lanes) to the center pump or pumps, is narrower than those of the two outer pumps. If the flow lanes were equal in width to the minimum single-pump sump suggested by Mr. Iversen, there would be no hydraulic advantage to this design as the pumps could be located side by side across the sump, provided there was ample clearance for the installation of the motors and related features. Since the pump columns or suction tubes for the echelon design of sump are not located directly downstream from other pumps, it is concluded that eddies or vortex trails introduced by upstream pumps would not influence the operation of downstream units.

A second design of pump intake is presented for consideration as shown in Fig. 12 of this discussion. Pumping plants usually are designed so that a straight flow of approach is obtained at the trashracks and, while it is realized that streamlined rack bars would be more efficient, rectangular bars are used for other considerations. Test data on the losses through rack bars of various cross sections are meager and this lack of information may explain why there is so little variation in trashrack designs.

The losses through a trashrack are dependent upon the bar shape, bar spacing, and the angle of approach of the stream flow to the trashrack. Head loss through a trashrack diminishes as the cross section of the rack bars become streamlined and as the spacing increases. The effect of the angle of approach α, is not the same for all bar cross sections. Rectangular bars cause increasing head loss as the angle of approach increases. Bars having a rounded nose, and square, round, or tapered trailing edge, and symmetrical airfoil-type bars cause a decreasing amount of head loss with increasing angle of approach until an angle of approximately

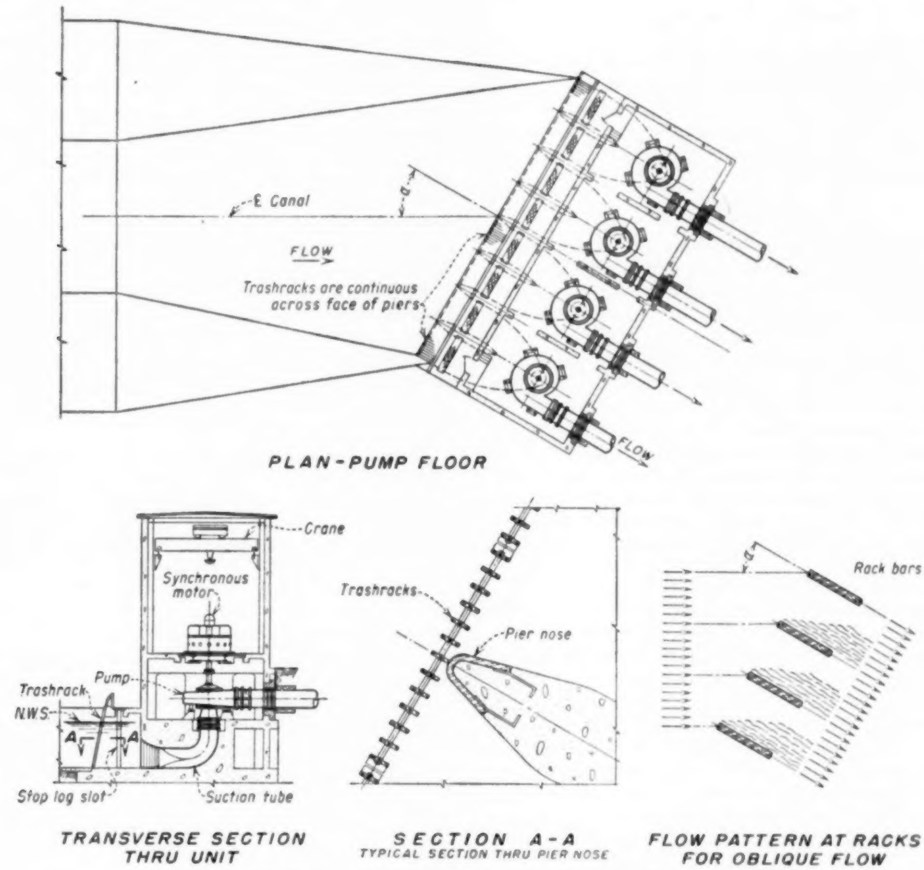

Fig. 12 Pumping Plant With Oblique Approach Channel

30 deg is reached for usual bar spacings. Rectangular bars cause a greater head loss than do the modified type of bars at all angles of approach. The length-to-width ratio of the rack-bar cross section also influences the head loss; a ratio of about 5 to 1 has been found to be the most efficient.

An efficient trashrack design would have the following features:

1 The trashrack would have the largest bar spacing that anticipated trash conditions and minimum opening in the pump impeller would permit.

2 The trashrack bars would have a rounded nose and semi-streamlined cross section.

3 The angle of approach would be 30 deg or less with the optimum approaching 30 deg.

It has been a general practice to slope the trashracks at an angle of 15 to 20 deg from the vertical. This practice was followed to facilitate cleaning and to present a pleasing appearance. From the efficiency standpoint a trashrack set normal to the direction of flow in the vertical plane will cause the least amount of head loss. Trashracks used for canal or river structures and for lakes or reservoirs which have a small fluctuation of water level usually are not deep enough to prevent the use of hand rakes or unguided mechanical rakes. For trashracks of this type the inclination of the racks aids in keeping the rakes in contact with the racks. Reservoirs behind high-head dams often have a surface fluctuation of 100 ft or more and the depth to which the trashracks extend precludes the use of hand rakes or unguided mechanical rakes. Guided mechanical rakes are a necessity and by the nature of their functioning no advantage is obtained by having sloping racks. The best design in regard to the slope of the rack would be to make it as nearly normal to the stream flow as possible and still keep the rake in intimate contact with the racks. Where conditions require a guided or captive-type rake, or where such a rake is economically justified, the trashrack should be set normal to the flow in the vertical plane.

There are reasons other than increased efficiency for setting the trashrack and pumping plant obliquely to the direction of stream flow. A pumping plant set obliquely to the stream flow will require a shorter narrower forebay than will a plant set normal to the stream flow. The saving possible through the reduction of cost of excavation and material for the forebay structure usually will more than justify the expense of a trashrack using a modified-type bar. This arrangement also has proved to be a convenient device for changing the alignment of canals, which results in simplicity of construction as well as some additional savings in cost.

The information on trashrack performance presented here has been available for a number of years but apparently has received little consideration, at least in this country. The information on which the foregoing discussion is based was obtained from a test[10]

[10] As noted in "Hydraulic Laboratory Practice," ASME publication, 1929, p. 461.

report by Dr. O. Kirochmes of the Technical University of Munich, Germany.

The efficiency and lower initial costs of plants using oblique setting of the pumps and orientation of the plant structure with racks set obliquely to the stream flow should justify their being considered in the layouts of new developments. The Bureau of Reclamation has developed and consistently used these principles in its design procedure.

Author's Closure

The author appreciates the comments and contributions of Messrs. Morgan, Rankin, and Wilderson to the discussion.

Mr. Morgan points out that modeling of the actual pumps in model studies more closely approximates the flow conditions in the prototype than does the use of siphons. We do not believe this can be completely justified as most pumps are designed for nonrotational flow into the suction at the point of maximum efficiency and do not produce an appreciable disturbance in the flow pattern of the intake channel. In the event the pumps operate at capacities considerably less than or in excess of the capacity at maximum efficiency then the author agrees that the prerotation of the flow entering the impeller would affect the channel flow pattern and siphons would not produce a similar disturbance. The use of model pumps is of value, however, to determine the variation in power requirements with rotational flows in the sump assuming the models are of sufficient size to measure accurately the input power.

The author agrees with Mr. Rankin that suction scoops are not required in tunnel flows of 3 fps or less and that in new installations the intake should be designed for velocities not in excess of 2 to 3 fps. Owing to plant expansion or to pumps operating at capacities greater than anticipated because of reduced heads, however, these recommended velocities are often exceeded. In such cases the use of suction scoops will often result in satisfactory pump operation without undertaking the expensive alterations necessary to increase the tunnel or channel size.

The sump dimensions shown in Figs. 1, 2, 3, and 4 of the paper are conservative and are based on relatively high suction bell velocities of 6 to 7 fps. With lower-suction bell velocities the sump dimensions expressed as a function of the bell diameter can be reduced. The author believes that sump data based upon suction bell diameters can be compared only when the suction bell velocities are equal.

The information contributed by Mr. Wilderson is of interest and adds considerably to the data already available to guide the designer of suction tunnels and channels. Of particular interest is the use of the trashrack bars to achieve an oblique flow from the main intake channel as illustrated in Fig. 12 of the discussion.

Ten Years' Progress in Management

Foreword

By LILLIAN M. GILBRETH,[1] MONTCLAIR, N. J.

THE first management progress report,[2] entitled "The Present State of the Art of Industrial Management," was presented by a committee of which James M. Dodge was chairman and L. P. Alford, secretary. The committee submitted questions on management to various experts, assembled their replies, and deduced from them the contents of the report. It presents what was being done, the advocates and opponents of scientific management, and the important emphases. These include not only techniques of production, but aspects of human relations, calling attention to the importance of skill, of motivation, and of attitudes. Mr. Dodge, a satisfied user of scientific management, was president of the Link Belt Company; most of the other members of the committee were also manufacturers. The report is supplemented by a list of the important papers on scientific management printed by this Society, sixteen—written by ten authors. There is also a minority report, signed by one member of the committee, stressing the art of management as more important than the science. Dr. Alford did his usual fine objective job as editor, and the reports evoked much worth-while discussion.

In 1919 Dr. Alford wrote what he considered a supplementary management report, called "The Status of Industrial Relations."[3] This is not considered one of the "Ten-Year" reports, but is important in that it stresses the increasing emphasis on human relations. "Principles, practice, and law" are all considered and the main topics of interest listed and discussed. A bibliography follows. Again, the discussion is excellent.

The 1922 report is entitled "Ten Year's Progress in Management"[4] and is written by Dr. Alford as author, with no committee assistance. He was recognized as the most fitting person to evaluate the decade. He stresses that progress has largely to do with the human element—adding to the 1912 criteria the motive of service, and concluding, "Management is the agency by which community, state, and nation shall endure." An appendix contains accounts of management societies in existence at the time. Engineering schools which have management courses are listed, as among the evidences that management activities have broadened. The discussion is as interesting as the report.

[1] President, Gilbreth, Inc. Hon. Mem., ASME.

[2] Majority Report of Sub-Committee on Administration, Trans. ASME, vol. 34, 1912, p. 1131.

[3] Prepared at the request of the Committee on Meetings and Programs, Trans. ASME, vol. 41, 1919, p. 163.

[4] Prepared by request and presented during Management Week, October 16–21, 1922, and at the Annual Meeting of that year, Trans. ASME, vol. 44, 1922, p. 1243.

Contributed by the Management Division and presented at the Annual Meeting, New York, N. Y., November 30–December 5, 1952, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

The 1932 report, also by Dr. Alford, presents steady progress.[5] This includes more effective technical procedure, and better human relations as well as wider public service, mentioning national planning; also more courses in engineering colleges. The discussion, though short, is stimulating.

The 1942 report[6] was planned by Dr. Alford, who was assembling his material when he died. It was completed under the sponsorship of a special committee of the Management Division, by the devoted effort of George Hagemann, the vice-chairman of this committee who had worked with Dr. Alford. It is dedicated to Dr. Alford, and includes summaries of the 1919, 1922, and 1932 reports. It covers sections on the various aspects of scientific management. These had been assigned to various authorities in the field whose articles were combined into a report that reflects the thinking of the decade.

The 1952 report follows the pattern of the 1942, but with more participation of the Management Division Committee who discussed the plan of the report, the fields of interest, the assignments, and the procedure, in detail. Various authorities in the field of management have been asked to present and evaluate what has happened in the decade, and the reader will see for himself the expansion of the field and the problems and prospects it presents.

On the whole we have a challenging picture. Scientific management has expanded in its geographical application and in its application not only to industry and business but to government, to agriculture, to the home, to libraries, to hospitals, and to work with the physically disabled. Technical adequacy has not diminished, though the continued need for research is evident. Emphasis on human relations has increased, with both courses and work carried on in physical, mental, emotional, and social adequacy, and with a new emphasis on the art of communication. Local, regional, and national groups are united in a National Management Council which is a part of the CIOS, the international management group. The programs of the nine International Management Congresses so far held, beginning with the 1924 Congress in Prague, offer evidence of the expanding field, as do preliminary plans for the program of the tenth Congress, to be held in Brazil in February, 1954. To be noted also are the closer relations of industrial engineering with industrial medicine, psychology, psychiatry, and so on. This is evidenced in curricula of colleges, in programs of management meetings, but especially in research, both fundamental and applied.

Our chief problem seems to be to enable more people to have work that will produce not only necessary goods, but satisfaction to them and to everyone. We look forward to a future of effort, and of service.

[5] "Ten Years' Progress in Management—1923–1932," contributed by the Management Division and presented at the Annual Meeting, Trans. ASME, vol. 55, 1933, Man. 55-2, p. 7.

[6] "Ten Years' Progress in Management," presented at the Annual Meeting 1942, Trans. ASME, vol. 65, 1943, p. 213.

Management Division

The Management Division in planning this 1952 report has tried to clearly continue the form and purpose of 1912, 1922, 1932, and 1942 reports. We believe that each paper represents an important area of management, and that the pattern of this report will be of assistance to our Division in preparing the 1962 report, which will record the first 50 years of Scientific Management.

We extend our sincere thanks and appreciation to the authors who have unselfishly contributed their knowledge and valuable time in preparing this report.

ARTHUR M. PERRIN, *Chairman,* Ten Years' Progress in Management Committee and *Editor* of Report
ERCOLE ROSA, JR., *Co-Editor*

The Theory of Organization and Management

By ROBERT T. LIVINGSTON,[1] NEW YORK, N. Y.

THIS paper is intended to set the framework for the study of progress in management from 1942 to 1952. Our main thesis submits that great though the progress has been in management mechanisms, much less has been achieved in developing a true science of management. It is suggested that what we have witnessed is largely a continuously branching specialization and intensification of detail with very little feedback into the basic stream of theory. It proposes a return to Taylor.

A progress report is intended to answer three principal questions:

1 Where are we?
2 How did we get there?
3 Where are we going?

The year 1952 is a particularly apt one in which to do all three for it is apparent that management today is faced with increasingly new and increasingly difficult problems. In many ways the situation is not unlike what it was in 1912, and it seems wise in this paper, rather than merely to catalog the many unusual and indeed dramatic occurrences of the past 10 years, to look back further and in rather fundamental terms assess where we are now, where we were then, and how we got from there to here. Perhaps in the light of these 40 years of experience, we can, in some measure, assess the probabilities of the future.

Management an Engineering Problem

Men have managed men since prehistory, yet even to this day the current operations of an enterprise continue to present new and different problems to the manager. By prevoyance, planning, and preparation the solution of many problems may be routinized and presolved but because of changes in technology, changes in the institutions and goals of society, because man, as Gillespie has it, is a purposive, we will never "solve" the problem of management. However, it is believed that it is possible to evolve a rational theory or system of management.

In the 1913 Transactions of this Society is a definition of "management" which today stands equally valid:

"Management is the art and science of preparing, organizing and directing human effort applied to control the forces and utilize the material of nature for the benefit of man."

Here is a description and a value system. The engineer is not to be a mere scientist seeking truth; he is dedicated to the benefit of man. And we hold with Taylor's definition of scientific management written 40 years ago:

"Scientific management, in its essence, consists of a certain philosophy which results..... in a combination of the four great underlying principles of Management:

"Development of a True Science.
Scientific Selection of the Worker.
His Scientific education and development.
Intimate, friendly and co-operative relations between management and men."

It is the premise of the paper that regardless of how much the three last principles have been applied, developed, and used, that far less progress has been made in the first and the most important of all, namely, "the development of a true science of management."

Our central theme is, that for the peace of the world to come, management, and the engineer especially, must accept an increasingly greater responsibility for the operation of our society. It is maintained that we have, or science will give us, assets and resources in plenty but that the really great potential of society is the creative abilities of the people. While the nuclear scientist has unlocked the hitherto untapped energies of the atom the great potentialities of man's brains, ingenuity, and genius remain only partially utilized.

Progress in Management. Progress is a difficult thing to assess, since progress is not synonymous with change. The scientific-management movement, which formally is only just over 40 years old, has had a deep and profound effect upon our society and there is no doubt that its importance grows daily. Its use is not restricted to capitalism; on the contrary, a recent, quite important book came out of Russia called "Industrial Management in the U.S.S.R." which is a very interesting history of how the Soviets have tried to take what they thought was good from scientific management and put it to their own use, always, of course claiming that Marx, Lenin, or Stalin originally set it forth.

Any categorization is necessarily subjective and is always an adjustment between the desire to simplify and the wish to specify. But in considering the question of management, and especially its history, one is forced to categorize in some manner. The four categories, which are set up in the following, differ from Taylor's but perhaps the 40 years will account for the difference:

1 Procedural and process evolution.
2 Change of attitude about the worker.
3 Evolution of a rational process of decision making.
4 Development of a science of management.

Number four was Taylor's No. 1. Number two will comprehend the three others. It is possible to say that procedural and process evolution was implicit in his thinking and finally that progress in management may best be measured in terms of the evolution of a rational process of decision making.

Management is the taking of action to answer stimuli; in a word, it is a solving of problems, problems which change in time. Thus it is maintained that "progress" in management can only be assessed intelligently in the light of the problems that management is called upon to solve. These problems are in the fabric of the social and economic structure, are indeed the dynamics of society.

Fields of Application. While engineers have contributed greatly to the development of a theory of management, as designers they organize, and as operators they administer, it is obvious that all managers are not engineers. There are many fields of management:

1 Public and Institutional: The government and public institutions such as the schools and hospitals.

2 Private Institutions: Industrial and business associations, labor, and other associations.

[1] Professor of Industrial Engineering, Columbia University. Mem. ASME.

While these are distinctly different as to the fields in which they operate and the goals they seek, yet there are certain common characteristics which make it possible to study them all together. The common denominator is that they all are dealing with men, in a goal-seeking association, expressed usually in some kind of a process dealing with some kind of material, existing with a social matrix, and interacting with other institutions while serving society.

The Climate Within Which Management Has and Must Function

The function of management is omnipresent. It exists in government as well as in industrial associations, and it exists in organized labor. However, while we maintain this, our accent in this paper is on the management of industrial enterprises. While there are many factors that influence management in so far as its decisions are concerned, we will first set forth six classes of influences:

1 The industrial institution known as the corporation.

2 Supplier of capital.

3 Labor, not as a supply but as a most important factor in production and society.

4 The State, Federal and local, as an institution.

5 The state of science and technology.

6 The social matrix—all those forces which determine the demand for the industrial institution's existence but also those forces which influence and make demands upon the managers.

The making of decisions is regulated by the location of major power or by the relative weakness of the contenders. Therefore much can be learned about management by studying the relative power positions of those who bring pressures to bear upon management.

Management makes decisions and attempts to implement them, decisions about the goals of the institution which it is operating. Since different institutions have different goals, so different kinds of management will make different kinds of decisions but the process of decision-making is universal. Decisions are made involving other people, men, machines, and process. Thus decisions are limited and are not free.

Throughout history there have always been rulers, people who possessed authority, made decisions, and exercised power. There has not always been calm acceptance of this power, and the early idea of ultimate and absolute power over all facets of life has been discarded completely in western civilization; in a word, power is divided. As power became divided and because so many of man's activities overlap, there is a natural conflict as to the relative division of this power.

The Industrial Association

Let us assume that an "industrial association" is an entity existing at a point of time in a force field which we will call the "social matrix." The industrial association is one kind of institution within that matrix—there are others.

Consider an association as an entity which has been formed in some manner. It interacts with the social matrix. It establishes routine channels of communication with other institutions, and it adjusts its internal structure, perhaps according to the principle of least effort. But certainly an association, at any instant, can be considered in a sort of "steady state" in the thermodynamic sense. It receives communications of various kinds, it carries on its process, it grows or decays—constantly readjusting within certain limits—and it emits communication.

The corporation is an important institution, worthy of far more thought than can be here devoted to it; it is old but its use for production is but little over 100 years old, and its public use almost exactly 100 years old. Its adoption was seized upon avidly and used widely and quite naturally there were abuses. As the abuses increased and the advantages gave unusual power to individuals and groups, quite naturally antithesis arose—the ICC, the Sherman Act, and so on.

The corporation of 1892 was predatory and exploitive; the corporation of 1952 is almost entirely concerned, as Peter Drucker has it, with self-perpetuation. It has, in effect, become a member of society with all the rights and responsibilities which accrue thereto.

The Supplier of Capital. Out of the same social matrix we shall abstract the supplier of capital as another directing force. Regardless of the kind of society, funds must flow, plants must be built, assets must be available. The problem then revolves about how this capital will be formed and the influence that the contributors of the capital will have upon the decision-making manager.

Time was when the venturer was the owner and usually the manager—but by the end of the nineteenth century there was already a distinct differentiation between the operative manager and the supplier of funds. The supplier of funds was a person who dealt in relatively large sums of money and management's decisions were slanted toward that person with money; ownership was important. The archtype of American Society was the successful individual entrepreneur.

Came the first World War and the "Baby Bond," followed by no-par-value stock. As more and more people went into the market, ownership became diffused until it was no longer an important individual controller. We had a new economic era; the American industrial corporation was no longer bound by the old economic laws. Employees owned stock, customers owned stock, everybody owned stock. Competition[2] was the great controller of our society. There were vague rumblings of discontent it is true, and there was a "farm problem," but finance and "Wall Street" were in the saddle.

Then 1929!! Since there had to be a scapegoat, the fingers pointed at the "banker." Banks were closed and at last money came to mean what it was designed to be, a medium of exchange, and the banker what he really is—a man performing a useful service for society. But who was to assume leadership? The banker was out, the industrialist was not yet ready to assume the onus—only the state remained, so, fumblingly at first but then with great confidence, government took over....the "New Deal," national planning, the good of society, the allocation of raw materials...As World War II came, government assumed an increasingly important part in our economy and was increasingly a determinant in management's ability to make decisions. Regulative and administrative agencies entered into all the many facets of industrial and corporate life.

Furthermore, with a war upon us, government became a big customer—in many cases, the largest single customer. As such, government was in a position to exercise pressures upon management in a new direction—it could regulate as government but it also could control as the single most important customer. Now the state began to assume the dominant position in decision-making that previously had been exercised by ownership, and the compass of management's valuation was turned from distribution of profits to the shareholders to operating within the limits of government restrictions.

Labor

Meanwhile labor, as an institution, has had its tides, its rises and falls. In actual fact, in the broad sweep, it was not until 1942 that there was any power-sharing with labor except in

[2] Except in the public-utility field where the industrial mergers of a generation ago were being duplicated.

individual cases. At the turn of the century management offered jobs on a take-it-or-leave-it basis, regarding labor very definitely as a commodity to be bought and used. If the domestic market would not supply the labor, then labor was imported. The Sherman Anti-Trust Act and the "Yellow Dog Contracts" tied up labor even more completely; under them, when the job was taken, it had to be kept and labor became even more of a commodity, bought and contracted for, with the contract regarded as a most sacred thing. The 1920's did not improve labor's position greatly although it did lay the ground for many of the later gains. This "era of good will" was a time of prosperity when wages could be raised without too much trouble since it is more important not to have a work stoppage. From 1932 to 1942 many of the problems of labor organization were worked out; the pattern of the relations of labor to government and to management were explored and the groundwork was laid for many events which have left labor, at the opening of 1952, in a new position—a third contender for power in society.

While the "top" management of labor learned and grew to power in a struggle for recognition and power, and quite naturally, thinks in terms of immediate goals and fundamental differences, the younger men have experienced a period in which their right was recognized. Their problems have been problems of adjustment to situations of permanency rather than of emergency. It is believed that the next 10 years will see a change in labor leadership of great importance which may well make possible great technological advances. As the unit cost of labor rises, management must increase productivity per labor unit and this can be done only by increased mechanization and increased participation by labor in labor and waste-saving. If the demands of labor for continuity of employment are to be met, then labor in turn must guarantee against stoppage of production while disputes (and there always will be disputes) are going on, but management in turn must learn to control production and this has many curious implications.

The State

The state is one of man's earliest institutions but the American State of 1952, or the monolithic Russian State is a far different thing than the police-power state of, say, 1892. Here is a complete change of thinking in 60 short years. The ambit of the state's activity has increased to a point that no industrial decision today can be made without consideration of the state. In the early days, regulation and control were largely ex post facto; most regulative agencies were local as indeed were most enterprises.

The property of municipal administration was finally recognized in the growth of the city manager. The over-all need for forethought was reflected in the growth of planning agencies at all levels and the need for control gave rise to the regulative agency.

At the same time, administration within the state was changing drastically, both in kind and degree. The change in quantity is well known but there has been a great change in character as well, a professionalization of public administration and an examination of quality.

The Social Matrix. Next there is a set of what engineers may call imponderables but which we here name the "social matrix." None of us, nor our institutions, exists in a vacuum. On the contrary, there is an "N" dimensional sea of social forces that impinge on all of us and upon our institutions, forces which create our system of value judgments. For example, what was accepted as merely shrewd in 1892, may be criminal today, and what was once supposed to be a local responsibility is today a federal charge.

This thinking permeates our whole life and is, in this particular case, reflected in the position of the manager. A generation ago the profession of public-relations councilor was unknown, and few people formally consulted public opinion. Today most companies have public-relations councilors and public polls are avidly followed.

Interpersonal Relations. In the "good olde days" there was a very intimate relation between the manager (who was the owner) and the worker. For better or worse, good or bad, it was a direct and personal one, and man, it seems, craves personal relations. It is probably psychologically better to be browbeaten and kicked than to be ignored. While prior to 1892 there were large aggregations of labor, such aggregations were the exception rather than the rule. However, as aggregations grew and this was not a rapid process, absentee ownership and professional management grew, and interpersonal relationships became increasingly impersonal.

True it was that there were many owners and managers who were humanitarians and who were interested in the "lot" of the worker, and who tried to "improve" matters—Pullman, Hershey, Kohler, and others, and it is a false stereotype to think of owners and managers deliberately "grinding the face of the worker into the ground." There were abuses but they were faults of omission rather than commission. Humanitarianism, however, did not, could not, will never work, because it is a gift from above as a favor and a favor may always (or, at least it is feared) be withdrawn. It is somewhat like the difference between the United States and the USSR Bill of Rights. That which is a "right of the people" is very different from that which is a "privilege" granted by a superior authority.

It has been said that there is a profound difference in the working force of 1952 and of the working forces of, say, 1942 and 1912, in large part resulting from the increasing educational age of the worker. This, in effect, means that there is a contraction of the difference of formal education between the manager and the person managed. This is important but can we say casually that the average increased educational age is advantageous? There are problems of adjustment and more additional years of formal education and especially modern education do not of necessity mean increasing ability to assume responsibility. The individual has greater rights and privileges but is that reflected in greater assumption of social responsibility? People today are increasingly aware of their rights but are they similarly aware of their responsibilities? This must be learned and the learning process is slow.

The State of Technology. A Newtonian physics yields very different results from a quantum theory and the difference between aristotelian and nonaristotelian thinking is even greater. But we, in our pseudosophistication must not forget that there is a lag between the knowing and the general acceptance of the knowing.

Changes occur only in answer to stimuli. A useful means or mechanism may exist unused until there occurs a "reason" for its adoption and continuance. It may be adopted in isolated cases because of efforts of one man and acceptance by another. It may continue in isolation if the pressures which cause it to be accepted are maintained but it will not gain widespread acceptance until and unless a "need" arises. By need is meant that its acceptance solves a recognized problem better than the way that problem is at present being solved.

Procedural and Process Evolution. It is in terms of the evolution in procedure and process that management progress has most commonly been considered. Procedure, that is to say, record keeping, and the like, and process: all those matters which are related to, or spring from, production engineering.

It is important to consider all the implications contained in the early Taylor work, for it included not only a small break with

tradition, but a rather complete break: The worker was to be trained to do a specific job, not a total or general skill, but a specific job. That job was to be analyzed and the tool redesigned as requisite.

The "art of cutting metal" is also equally provocative, for here is an illustration of solving a problem by a head-on attack. This had implications far beyond its particular field and may indeed be considered the basis to take off from the building of a rational theory of decision-making.

The work of the Gilbreths, though independent, was a natural development. The job was still further analyzed into basic components (described as therbligs), the work place was studied, the positioning of the work was considered so that the worker could be taught to do the job in the most effective way.

From time and motion study to "simo-motion" chart and process charts, from work analysis to plant layout and materials handling, inventory control, location of toolrooms and economic lot size, the derivation is obvious. Routing, scheduling, and dispatching, man and machine loading, and the assembly line, lead inevitably to the completely automatic factory which is now a possibility.

The Streams of Management Theory and Their Contributions

Management has been said to consist of two related functions, "organization" which is "design," and "administration" which is "operation." The history of management may be viewed usefully from the point of view of the various schools of thought which have considered either the problem of organization or of administration. The problem of management for the future is to adapt the findings of these schools to practice and to set problems for these schools to study. In a word, it is said that research is essential and that it is the duty of modern management to encourage, aid, and abet research.

We will study the problem in terms of the following:

1 Knowledge of individual behavior.
2 The theory of groups.
3 The theory of communication.
4 A rational basis of decision-making.

It is important, however, to realize that in any such approach we are abstracting from the whole. Any study must comprehend the "gestalt"—the physiological, the biological, the psychological, the sociological, the anthropological man, and indeed the spiritual man. It must not be forgotten that he is simultaneously all of these. In a somewhat similar manner while we have arbitrarily, perhaps, divided this section into four parts it must not be forgotten that they are all parts of a greater whole.

While the origin of much of the formal thinking about management was undoubtedly in the engineering field, yet management basically deals with human beings. Not unnaturally the psychologists have made important contributions.

Munsterberg's writings went largely unrecognized. It was the psychological tests of World War I that first gained recognition for the psychologist in the field of management. Here for the first time were tests applied to a large adult population. Management, always avid for help, believed that psychological tests were the answer to all their problems. Tests were applied widely; the round peg was fitted to the round hole. Unfortunately, it was found that both peg and hole reacted to each other and to time, and the perfectly fitting peg today will not necessarily fit tomorrow.

But psychology had much more to offer. Gillespie in England set forth that the task of management was to deal with inerts, dynamics, and purposives, by which he meant materials, machines, and men. He pointed out that men had aims, ideals, needs, and wants apart from the work association and that conflict and frustration arose within the individual which would be reflected in his work attitude. Lecky, a Columbia psychologist, while working for the Long Island Lighting Company, set forth a useful similar theory—the theory of self-consistency. This theory said that each individual was the center of his own particular universe and that all of life had to be consistent with his concept. In a word, Lecky explained the remark so often made in exasperation, "Who does he think he is?" for Lecky maintained that a person behaves in the manner of the person he thinks himself to be.

But it is not alone from the psychologist that insights are to be gained. The biologist and especially the mathematician-biologist, Rasheffsky at Chicago, and Von Bertelanfy at Toronto, seem to have much to offer. The biologist, of necessity, has developed considerable skill at taxonomy and is concerned with organization and adjustment.

Group Theory. Important though as understanding of the individual may be, industry is as much concerned with the behavior of people in groups. Basically this is the field of sociology but it is also the field of anthropology. It is only recently that the sociologist has moved out of studies of the family and similar institutions and that the anthropologist ran out of island and isolated cultures, and that both of them have looked at man in the work situation. The engineer has been doing the same for years. But the sociologist and the anthropologist bring new techniques and new ways of thinking to bear on an old problem. There is no doubt that an understanding of their theories can be of great importance to man charged with the responsibility of organizing and administering.

Group Dynamics. Group dynamics is intimately associated with the name of Kurt Lewin. In this country, after sampling many climates, he settled at the Massachusetts Institute of Technology, but his followers split up and now Michigan as well as M.I.T. inherit the mantle. Lewin's concept was a "field theory." The importance of his concepts are in the suggestions rather than in the concepts themselves. Lewin's topology and Bavelas' development of hodology remain nonoperational but highly suggestive.

Among others the sociologists have another school which for want of a better name may be called the "psychosociometric" school. What is, is, they say, and as you are so will you behave. Moreno and his group say, in effect, that a person's likes and dislikes are important, that you can measure the success of an association (or an organization) in terms of the satisfaction that the person within the association gets. They measure this in terms of whom do you like, whom do you talk with, whom do you like to be with, and so on. The psychodramatist's concept is based upon some hitherto unrecognized inspiration of Emma Sheriden Fry, a great actress of a generation ago.

Shartle, on the other hand, is distinctly different. His statistical mechanisms lead to the consideration of many factors—factor analysis—and the production of profiles. Will this man fit this job and how well? A courageous attempt to predict in a field where many doubt if it is possible.

Communication

When history has finally been written it may well be that the present period will be known as the period of "communication theory." It is important to realize what communication really means. It does not, for example, merely mean the ability to interchange messages—it is much deeper and much more important than that. It means the ability to interchange information and perhaps in the end the religious idea of "being in communion." At least, it has been recognized that most opposition

arises, not from fundamental differences, but because people do not understand one another. Perhaps the first in this area are Mayo, Roethlisberger, and, in general, the Harvard School of Business Administration which, by the case system, first investigated such individual situations. In spite of the obviously justifiable criticism of the case system there is no doubt of the great contribution which the school has made.

The case method has been followed by the interaction theory so ably set forth by Arensberg, Bales, Chapple, Homans, and Whyte. Here, with the exception of Bales and Homans, is an attempt to measure, without content, the interaction of people, of people in groups, and to educe scientific laws therefrom. Bales, with a system of twelve categories, attempts to measure intent in part as well as interaction.

The sociopsychologists and the psychiatrists by means of depth interviews, critical situation analysis, and similar techniques, attempt to get behind the obvious interactions that the interactionists measure and explain as well as express what is going on.

A Theory of Decisioning. It is the thesis of this paper that, in the end, the progress of management will be traced in terms of the emergence of a rational basis of decision-making.

In this aspect we can recognize a series of sequential and temporal steps in the history of management:

(*a*) Importance of use of facts and data.
(*b*) Rise of early statistical and graphical mechanisms.
(*c*) Idea of partial goal setting and realization.
(*d*) Control of quality, as well as quantity.
(*e*) Information and communication theory, feedback, and cybernetics.
(*f*) Operational analysis.
(*g*) Process of decisioning.

(*a*) The 1912 report spent much time discussing the importance of making decisions based on facts and data rather than on hunch and judgments. Also, the idea of both internal and external facts: that means comparison. It seems hard to believe today that that should have needed to be discussed.

(*b*) It was the interim report of 1919 that first widely introduced to engineers the statistical and graphical methods. Of course part of the work of Gant was graphical. However, the need of statistics arose as soon as the desirability of using data and facts was accepted. The more facts and data were collected the less the human mind could comprehend, evaluate, and understand them. Statistics serve two major purposes:

1 To reduce a mass of data to the number of numbers for which a person can comprehend the interrelations—five perhaps conveniently, and these five are

$$N, \bar{X}, \sigma, \beta_1, \beta_2$$

2 To discern interrelations—that is

(*a*) Curve fitting $y = \phi X$
(*b*) Correlating $y = \phi\,(XYZ\ldots\ldots.)$

(*c*) The first World War impressed upon management the concept of goal setting in time and the idea of partial goals and control; that is to say, measuring partial accomplishment versus partial goal and the necessity of expediting.

(*d*) *Quality control:* While there was physical interchangeability during World War I, it was not as good as needed. Complete inspection schemes were expensive and some inspection is necessarily destructive. Between 1924 and 1932, Shewhart and others developed quality control and sampling. This was an extension of the control theory in part, but it was also a bridge to the concept of decision-making. Statistical quality control inserted uncertainty into inspection—risk; consumers' risk and producers' risk. Before it had been a rather aristotelian process—you knew or you didn't know. Now it was possible to evaluate the state of knowing or of not knowing, and to make decisions as to (*a*) rejection or acceptance; (*b*) changing or not changing.

(*e*) *Information and communication theory; feedback and cybernetics:* Norbert Weiner and his cybernetics give us promise of the automatic factory which now is practical as well as possible, and Shannon and Weaver in the field of electrical communication have given us many useful ideas that will soon be embraced by the emergent theory of management. It may well be said that the future will not be so much concerned with gathering new data as it will be concerned with judging the pertinence and relevance of the information that it has available, its accuracy, and its usefulness.

(*f*) *Operations research:* Operations research is a relatively new type of application of scientific thinking to certain kinds of management problems. It was developed extensively for quantitative study of military operations in the British and American forces during World War II. Obviously, not limited to the military, it is strongly analogous to the best work in scientific management. (For example, a good deal of Taylor's studies could be classified under this heading.)

This development, in which scientists of many disciplines contributed to the solution of many complex military problems, is now being extended to industry. Advanced techniques of quantitative analysis along with probability theory and statistics are being used to obtain answers to production, organization, and control problems. . For example, the optimum number of maintenance personnel in a large transportation industry, the control of sales in a retailing operation, the development of sampling techniques in accounting are among the areas in which scientific teamwork has been found to be fruitful. Development of new measurements of effectiveness in many fields such as scheduling optimal utilization of plant facilities, routing and control of traffic, or the conduct of a sales campaign can provide managers with a sound quantitative basis for decision. In such roles, operations research is an important achievement of the last decade and is destined to leave a profound influence on scientific management of the future.

(*g*) *Process of decisioning:* If we expand the concept of decision-making to include, on the one hand, the process by which the decision is arrived at, and on the other hand, to include the process by which we implement or make the decision "work," and if we further recognize that this is a continuing, dynamic process rather than an occasional event, then decisioning means something quite different than heretofore and becomes the basis of all managerial action.

The Future of Management 1952–1962

The year 1962 will mark the golden jubilee of the scientific-management movement. What progress will be noted as we look back toward 1912, to what extent will we have moved from what was set forth in Taylor's monumental work? To what extent will we be able to say we have extended the concepts of those pioneers who founded the first management society? It is important that we be able to say that we have progressed, for the real problem of tomorrow, as it is of today, is "management." We must develop our knowledge of management for in the end we must apply the principles of scientific management to the world or we will at least retrogress, or perish at the worst.

There will be a wider understanding of the fact that any association is dynamic and must be in a constant state of change. That is to say, the function of routine production will become widely differentiated, as it is in some companies, from the function of organization and analysis. It will be recognized increasingly that

here is a continuous problem which requires continuous attention. The problem of long-time self-preservation is not identical with the problem of short-time profit-seeking. It is believed that these 10 years ahead will see the development of a valid, operationally verifiable theory of management. There will have been a sufficient number of experimental tests in "on-going" situations to demonstrate its great potentialities and there will be widespread acceptance that management is, in part at least, a science, which will in no way deny that there may be an administrative art. Science may never produce genius but it will help to avoid the little errors and mistakes which by compounding one another can build up into catastrophe.

Organization, as Alvin Brown understands it, will come into its own but not in a mechanistic sense. Synthesis will show how the findings of the social scientists, the anthropologists, the sociologists, and psychologists are related to, and modify the purely mechanistic approach of Gracuinas and Davis, and research will discover the functional form, as well as the design factors which will be used in producing more flexible productive structures.

The function of management, as suggested nearly a generation ago by Burnan, will become widely recognized but on a somewhat different level. The manager will increasingly become a catalyst, rather than a carrier of authority, a problem stater rather than a problem solver. Not so much because the problems are of increasingly greater difficulty, or because of their complexity, but because of the sheer number of people who will be involved by a single decision, and the need for their acceptance of the decision. Thus a decision accepted by a group does not need implementation and while the decision theoretically may not be as good as another, yet its acceptance may make it, on the one hand, more effective, and on the other hand, may build a bridge of acceptance to the total goal at a later date.

The average executive of today has far too much information—he knows too much about too many things and his time is far too much occupied with the consideration of unimportant things—"administrivia" as some unknown genius has phrased it. It is forecast that the next area of advancement will be in the evaluation of the pertinence and relevance of information, a field which is almost virgin. How much information is required to make a "good decision?" That, of course, depends upon the importance of the consequences of the decision. It is conceivable that a single piece of information might contain, say, 80 per cent of the relevant data and that a decision based upon that single piece of information might have a probability of being 0.8 successful. As additional information is received it does not follow that the decision made on the basis of the additional information will be successively better, but it does follow that the decision will be successively delayed, and that may be important. Thus an 80 per cent decision made today may be better than an 85 per cent decision made a week from today. All of this seems important but it is a matter which cannot be explored within the framework of this progress report but which is of great importance in so far as a theory of management is concerned.

We will increasingly use the available knowledge of the individual and of the group as our knowledge of communication and information theory grow, and most particularly we will gradually learn how to release the creative ability of the individual—the great potential of the future.

Conclusion

This paper appears under the name of Robert Teviot Livingston, the executive officer of the Department of Industrial Engineering at Columbia University, but a report such as this is never written by one man. In actual fact, every member of the staff of the Department made some distinct contribution, as, indeed, did the students. More especially is credit to be attributed to a research team under the direction of the author's graduate assistant—Mrs. Gene Weeks. This team consisted of Mr. Benedict Wengler, Sociology, Mr. William S. Sachs, Economics, Mr. Richard S. Moore, Public Administration, who collected a mass of material from which this report is abstracted. My thanks are due them, and it is freely and gratefully offered.

In closing, however, may the author add that he feels very humble in the attempt to do justice, in these few words, to what is, he believes, one of the important movements of our modern world. The Greeks had a word for almost anything and while inadequate is not a Greek word, we will close on that accent.

Statistical Quality Control

By ELLIS R. OTT,[1] NEW BRUNSWICK, N. J.

Introduction

These last 10 years have been very important ones in the development of statistical quality control. In 1942 there were only a few men who realized that this new science could provide a major contribution to industrial operations. There is now overwhelming evidence that their expectations were completely justified. Statistical quality control has been established as applicable in essentially every industry: to both processes and products; to both research and factory problems; to both persons and machines. This accumulation of experience and the acceptance by industry of the importance of this science are outstanding developments of the last 10 years.

Statistical quality control has many different applications some of which are important in one field and less important in another. But there is now a substantial body of knowledge, techniques and philosophy forming the core of a program of quality control. Techniques of statistical control have been applied during this decade to many important phases of industry: to the guidance of production processes; to inspection operations; to factory experimentation and trouble shooting; to the design of experiments and the analysis of experimental data; to the study of consumer wants; and to the aid of management in its top-flight decisions. It is recognized that these programs are related and that management must provide for an integration of these many functions.

According to the Bureau of Ordnance, Department of the Navy: "Statistical quality control is a major contribution to manufacturing efficiency..... Altogether, statistical quality control is becoming recognized both in Government and private industrial plants as the hallmark of efficient management....."

What Quality?

We talk about quality control but what quality do we plan to control? If we say, "The quality of the product we produce" then this is only a fraction of the answer, at least until it is very broadly interpreted. We prefer to include many (economically important) factors not always thought of in connection with

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quality. For example, the quality of a machine or operator performance is improved if the amount of rework is reduced; if the scrap pile dwindles; if the down time is decreased. These are subject to the methods of quality control. It is fortuitous that these economically important developments usually go hand in hand with improved quality in the ordinary sense.

It is important, too, to realize that "quality" is a function of two things—the design, and the conformance of units to that design. The contributions of quality control relate to both of these items, but the simplest and often its principal contribution has been the conformance aspect.

Management has found that many of the economic advantages of quality control have been attained within the design of the product or operation. They have been attained by foremen, engineers, chemists, and other specialists using statistical and other techniques—but they have been attained most effectively as a project based on the teamwork of representatives of several groups working together.

Functions of Quality Control

Management has learned that there are many important parts of a modern program of quality control. It is not essential that these functions be set up in a department labeled "Quality Control." It is important, however, that there be provision made for a co-ordination of the following functions:

Consumer Wants. An analysis of consumer wants related not only to the type of product but to the performance of that product. It is not easy to get a reliable prediction of consumer wants—but evidently it is important to have one before the product is manufactured.

Research and Development. Designed experiments in research and development, or production, are being accepted as an important phase of a quality-control program. Experiments are being designed which provide for the simultaneous variation of two and more factors in an experiment. Recently developed methods of experimentation as well as the older method of analysis of variance provide several times as much information as that obtained by varying one factor at a time.

Qualitative Factors. Management realizes that skilled operators, good machines, adequate lighting, friendly employee-employer relations, and a host of similar factors are conducive to good quality. But these factors alone will not insure quality.

Control-Chart Techniques. These are the effective techniques developed by Dr. Walter A. Shewhart (Bell Telephone Laboratories). Management is accepting these techniques as an important guide to production operations. A better-quality product can be assured, and can be produced more economically, by guiding the process than can be obtained by relying upon inspection.

Inspection of Materials and Parts. At many stages throughout the manufacturing process there is a very important role being played in inspection by acceptance-sampling plans. Sometimes these plans replace 100 per cent inspections; sometimes they are introduced to provide a needed check on materials or parts before they are moved from one department to the next.

Any of these inspection operations can perform a more constructive function than the traditional one of separating the "sheep from the goats." They must supply constructive information to the production operators.

Analysis of Consumer Reactions. Is the customer satisfied with the product or service? Even when the answer is "yes," it is not appropriate to relax. Competitors are alert. Is it time to start the cycle all over again? This program becomes a dynamic one, as this cycle is repeated again and again.

It is significant that quality control provides us with a quantitative basis for a decision in these problems of management and science.

Dr. P. S. Olmstead[2] (Bell Telephone Laboratories) indicated the many-sidedness of a quality-control program when he wrote (1945): "...(the) statistical method in so far as it applies to sampling consumer wants, research and development, design, specification including the setting of tolerance limits, inspection, and operational research to determine that standards are satisfactory, adequate, dependable, and economic. Obviously, all of these come under the general problem of quality control in industry."

A Decade of Developments

There were very few applications of modern quality control in 1942. Half a dozen men at the Bell Telephone Laboratories had developed techniques and philosophy; the Office of the Chief of Ordnance had scheduled training conferences for its different ordnance districts; there were about three books on the subject; and there was a small handful of industrial companies making cautious experiments. The use of control-chart techniques and acceptance sampling plans is synonomous with quality control in hundreds of plants today. Their use was essentially unknown 10 years ago.

Three areas of activity have been primarily responsible for this decade of development—training programs, technical publications, and the founding and ensuing growth of the American Society for Quality Control.

Beginning of the Quality-Control Movement

The evident importance of quality control in a possible war effort made it urgent (in 1942) that industries be given a chance to learn about it and be taught its methods.

The first university course in quality control was taught at Stanford University in 1941, by Profs. E. L. Grant, H. Working, and W. E. Deming. It was proposed that a series of similar short training programs could accomplish the desired training purpose by having a small corps of men offer such a program in key cities around the United States. Their purpose was to train both practitioners and teachers. In the period between July 17, 1942, and May 24, 1945, a total of 34 intensive courses in Statistical Quality Control were given in the United States. They were tuition-free and were financed chiefly or wholly by ESMWT of the U.S. Office of Education. Assistance was given by the Office of Production, Research and Development (OPRD) of the War Production Board. All were given in co-operation with collegiate institutions. Dr. Working[3] and Dr. E. G. Olds[4] of Carnegie Institute of Technology were in charge of programs.

During the period indicated, instruction was given to 1885 persons employed by some 600 companies. They were an effective means of teaching basic techniques and philosophy and they provided a real stimulus to industrial applications. To continue the instructions given during these intensive programs (all but three of them were given on eight consecutive days) a series of five meetings was scheduled at each training center with

[2] Book review of "Statistical Methods in Industry," by L. H. C. Tippett, London (Statistician to the British Cotton Industry Research Association), Iron and Steel Industrial Research Council, British Iron and Steel Federation, 1943, published in *American Statistical Association Journal*, vol. 40, 1945, pp. 408-409.

[3] "Statistical Quality Control in War Production," by H. Working, *Journal of the American Statistical Association*, vol. 40, 1945, pp. 425-447.

[4] "Organizations Concerned With Statistical Quality Control," by H. Working and E. G. Olds, Report No. 2, June, 1945, Quality Control Reports, OPRD, Quality Control Program, Carnegie Institute of Technology.

the following avowed purposes: Of sharing experiences, to learn more theory, to discuss practical plant problems, including ways of "convincing" associates and management. Following these scheduled reunions, it was decided by most groups that they should organize a local society, to continue similar discussions. By February, 1946, there were 17 such societies in the United States with a total reported membership of 1186.

The American Society for Quality Control

It became apparent that the activities of these local sections should be co-ordinated. Leaders of the seventeen sections met in New York City on February 16, 1946, to form a national society. It was named the American Society for Quality Control (ASQC).

The society has had a steady growth from a reported membership of 1186 on February 16, 1946, to a membership of 6078 in May, 1952. The number of sections has increased from 17 to 59 during this same period, and now includes three sections in Canada and one in Mexico.

Many members of the American Society for Quality Control also belong to other engineering societies such as The American Society of Mechanical Engineers (ASME), American Institute of Electrical Engineers (AIEE), American Society for Testing Materials (ASTM), Institute of Radio Engineers (IRE), and the American Chemical Society (ACS) as well as those devoted to statistics and management.

The American Society for Quality Control became an Affiliated Society in the American Association for the Advancement of Science (AAAS), assigned to the Engineering Section (M), in December, 1951.

The American Society for Quality Control has set up, within its own organization, technical committees to consider particular problems in the following areas: Aircraft, automotive, chemical, electronics, textile, and standards.

Training in Quality Control (1952)

It is important that training be given at several different levels if a major program of quality control is to be established. At each of these levels—whether for the man on the bench or the plant manager at his desk—it is important to teach certain basic concepts which are independent of any complicated techniques. Yet these basic concepts have not always been well taught, and programs have sometimes failed or faltered as a consequence. Sets of demonstration models and gadgets[5] have been developed as visual aids for the teaching of these basic concepts. Their use has been singularly successful. This method of teaching is quite different from earlier methods in which the mathematical theory behind the techniques was often emphasized.

The following principles[6] are suggested in teaching basic concepts of quality control:

1 Keep it simple.
2 Keep it visual.
3 Avoid statistical terminology.
4 Keep to actual plant problems.
5 Keep it rolling.

These principles were formulated primarily for training in industry but they are applicable at every level of training. Evidently there are going to be "professionals" in quality control who must be taught adequate statistical techniques. However, this training should not be confused with the program of training these basic concepts.

In-Plant Training. A successful program of quality control requires the co-operation of many different groups within a company. The first steps in securing their co-operation is to give them an understanding of the fundamental purposes of the program. It is desirable to include representatives from engineering, research, manufacturing, sales, and purchasing in the same training group. These training programs can provide a real stimulus to a quality-control program. Then it becomes important that there be leadership to keep the program "rolling." This leadership must understand the many different phases of a quality-control program.

The major responsibility for training personnel has been accepted by our colleges and universities. Their programs have been of the following kinds.

Short Courses. The training pattern of the original 8 to 10-day (given consecutively) ESMWT courses has been continued in some colleges and universities. These intensive programs provide an opportunity for persons to attend from a radius of several hundred miles.

Evening Courses for Employed Personnel. In the larger population centers, travel is not a major problem. Consequently, local universities and local sections of the American Society for Quality Control have sponsored evening courses. These courses extend over 3 or 4 months, meeting one or two nights a week. Probably the first of these evening courses was the one taught by Dr. Paul S. Olmstead (Bell Telephone Laboratories) at Stevens Institute of Technology in 1939. These courses are a delight to teach. By promoting plant projects from each student—projects which proceed with the classroom theory from week to week—a work-study program is established. It isn't long before it becomes apparent that we are actually doing quality control—not just teaching it.

Even better opportunities are available in those few centers which offer a sequence of two, three, or more of these courses. There are many phases to quality control, and only a beginning can be made in one course. These sequences have had a most receptive response from industry. It is significant that the demand for advanced programs in this new field has come primarily from men already employed. They have recognized the value of combining the methods of quality control with their technical know-how.

College Courses for Undergraduate Engineers. Only a few of our colleges offer courses in applied statistics or quality control. Those which are offered are usually on an elective basis, although some colleges do require a course in applied statistics for students in industrial engineering.[7] Several engineering societies promote statistical training for its members, and it is reasonable to predict that colleges will introduce such training into their engineering curricula.[7] Some of our agricultural colleges have developed programs in applied statistics which include modern methods of experimentation and analysis of data. However, college training, either at the undergraduate or the graduate level, is not yet a major factor in the development of quality-control programs.

The Problem—1942

Although there have been many developments in our applications of quality control since Dr. Shewhart wrote his 1942 ASME report, his remarks in that report are as pertinent today as they were then. We quote from his report:

"An adequate science of control for management should take

[5] "Basic Concepts of Statistical Quality Control," by E. R. Ott and P. C. Clifford, Proceedings of the Rutgers Conference on Quality Control, September, 1951.

[6] "Using Training Conferences as a Quality Control Catalyst," by P. C. Clifford, Transactions of the Sixth Annual Convention of the American Society for Quality Control, 1952, pp. 35–40.

[7] "The Need for Statistical Quality Control in Engineering Education," by E. G. Olds, Transactions of the Sixth Annual Convention of the American Society for Quality Control, 1952, pp. 65–72.

into account the fact that measurements of phenomena in both social and natural science for the most part obey neither deterministic nor statistical laws, until assignable causes of variability have been found and removed......

"The steps involved in attaining and making the most efficient use of a given degree of control often involve the co-ordinated effort of literally thousands of employees, including physicists, chemists, sales agents, purchasing agents, lawyers, and economists.....many of them must be sold on the use of statistical control techniques if the control statistician is to have an opportunity of making his full contribution to management in the solution of its problems.....

"In the future, the control statistician must do more than simply study and measure the effects of existing cause systems; he must help his colleagues devise means for modifying these cause systems in the best way to satisfy human wants...he must help change that demand (for goods) by showing...among other things, how to improve the quality of these goods to the consumer. He must not be content with measuring production costs; he must help decrease production costs.

"The future contribution of the statistical control statistician lies not so much in analyzing data put to him as in helping to get data in which assignable causes have been segregated so that analysis will lead to valid conclusions not otherwise possible... There is also great need for creating, through college training, a statistically minded new generation of those natural and social scientists who will have charge of preparing, organizing, and directing the effort of those who are to control the forces and to utilize the materials of nature for the benefit of man."

The Problem—1952

Only in small measure have the goals, as stated by Dr. Shewhart in 1942, been realized. However, it is satisfying to realize that these goals are now accepted by most leaders in the field. This is a distinct advance and offers assurance of their eventual attainment.

The next steps involved in attaining these goals are primarily of an educational nature; instructors must be developed, many practitioners must be trained, and many more managements made aware of the benefits of a modern program of quality control.

Instructors must be developed who can combine an understanding of the underlying philosophy with the application of the fundamental techniques. It is more difficult to teach "where" to apply the technique, than "how" to apply it. There can be little expansion of any program of quality-control education until these instructors have been trained both in theory and in practice.

The demand for courses in quality control has been growing during the last 10 years. A few of our engineering colleges have offered some training, usually on an elective basis. It has been well received. It is important that we make available to many more students this supplement to their kit of scientific and management tools. Where successful programs are offered in undergraduate schools of engineering, similar programs have been requested by departments of the physical and biological sciences and agriculture, and by schools of business administration for their students in marketing, management, and economics.

It can be predicted that the application of quality control will increase manyfold within this next decade. There will bean increased use of different aspects of the programs within companies; furthermore, there will be an adoption of the program by thousands of companies not now familiar with any significant phase of a quality-control program.

References

The following groups of publications have had important influence on the development of the science of quality control:

1 The publications of Dr. Walter A. Shewhart, including his books:

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"Statistical Methods From the Viewpoint of Quality Control," U. S. Department of Agriculture, 1939.

2 "Industrial Quality Control," the official bi-monthly publication of the American Society for Quality Control. The first editor was its founder, Dr. M. A. Brumbaugh.

3 Acceptance sampling:

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4 Four publications which have had a strong influence:

"An Engineers' Manual of Statistical Methods," by L. E. Simon, John Wiley & Sons, Inc., New York, N. Y., 1941.

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"Statistical Quality Control," by E. L. Grant, McGraw-Hill Book Company, Inc., New York, N. Y., 1946.

5 Research and Development:

"Statistical Methods for Research Workers," by R. A. Fisher, Oliver & Boyd, Ltd., London, England, 1925, and subsequent revisions.

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6 Management of Quality Control:

"Management of Inspection and Quality Control," by J. M. Juran, Harper & Brothers, New York, N. Y., 1945.

"Organizing for Quality and Waste Control," by E. H. MacNiece, Section V, Book 4 of Reading Course in Executive Technique, Funk & Wagnalls, New York, N. Y., 1948.

7 Elementary Mathematical Statistics:

"Elementary Statistical Analysis," by S. S. Wilks, Princeton University Press, Princeton, N. J., 1948.

"Introduction to Mathematical Statistics," by P. G. Hoel, John Wiley & Sons, Inc., New York, N. Y., 1947.

8 Many books have been published during these last five years. There is a place for others, such as the following:

(*a*) Applications of quality-control methods to the problems of management. The book, "Methods of Operations Research," by G. E. Kimball and P. M. Morse, John Wiley & Sons, Inc., New York, N. Y., covers this in part.

(*b*) An elementary (one-semester) text that will excite the interest of undergraduate students in science and engineering.

(*c*) An integration of experimental techniques and quality-control methods of analysis.

(*d*) A reslanting of the methods which recently have been developed in the social sciences for use in the natural sciences and manufacturing. See, for example, "Some Thoughts on What the Natural Scientist Needs From the Social Scientist," by P. S. Olmstead, in *Philosophy of Science*, April 1948.

Production Planning and Control

By EUGENE H. MacNIECE,[1] NEW BRUNSWICK, N. J.

ALTHOUGH many of the principles of production planning and control were developed years ago by Henry L. Gantt and others, they were not applied generally while industrial executives could still use rules of thumb based upon intimate knowledge of operations and applied with almost infallible judgment. In the course of industrial progress, the increasing speed, volume, and complexity resulting from applied science has demanded that management itself apply scientific methods. The accelerated rate of progress during the past decade necessitated broader application of already known methods for effective production planning and control. It also stimulated the development of some new concepts and treatment of production problems and a fuller realization of the importance of forecasting, planning, and control that achieves a balanced distribution of its benefits to the several segments of society.

Sales Forecasting

Prior to the past decade, sales forecasting was recognized as an essential requirement for effective production planning. Several industrial companies had achieved notable success in refining their sales-forecasting techniques to the extent that accuracy of these estimates provided a sound basis for the planning of operations as well as finances.

Despite the many dislocations caused by wartime demands during the past decade, sales forecasting was more broadly applied throughout American industry and commerce. The large majority of companies manufacturing goods to stock have accepted sales forecasts as being indispensable in development of their production schedules, procurement plans, operating budgets, personnel schedules, schedules of income, and plans for expanded production facilities. Most companies who manufacture to customer orders also have learned to use sales forecasts. Estimates of exact amounts of each specific sales item are not possible in this type of production but some estimate of dollars of sales or tonnage is used as the basis for operating budgets. The extension and refinement of predictive skills in the estimation of sales have provided foundations upon which plans for production may rest and adjustments may be made. As the accuracy of these approximations increases, however, adjustments are reduced and production planning is simplified.

Manufacturing Planning

In addition to accurate sales estimates, successful production planning depends upon effective and predictive manufacturing planning. Planners need to know the time required to perform each manufacturing operation in order to calculate machine capacities and requirements, to determine personnel requirements, and to develop detailed departmental production schedules. Prior to the past decade, only a few companies had assembled sufficient basic time data to predict equipment and personnel requirements needed for the production of various quantities of new products. Such means of prediction were, moreover, limited to industrial operations similar to those within the pattern of existing operations. This was so because the time data were basic for certain categories of operations but not sufficiently fundamental for all kinds of industrial operations. This meant that in the large majority of cases preliminary planning took the form of first approximations in which operations and operation times were rough estimates. Actual operations had to begin before final smooth planning could be based on time studies of them. Under such arrangements, planning usually followed operations with a time lag of several months and sometimes years. As a result, true costs and outputs were never known in advance, thus increasing risks of failure and never permitting replanning or abandoning a product or enterprises showing unfavorable promise of profit.

During the past decade, studies that had been taking form in previous decades were stimulated by men who should be called industrial scientists because of their scientific approach to the problem of developing fundamental time data having almost universal application to industrial operations. These developments, within their realm, are as essential and far-reaching as those that prevail in the natural sciences.

Prediction of the time required for part of some operations, particularly in metal cutting, go back to the time of Frederick W. Taylor. The machine cycle time, based on feed-and-speed information could be predicted accurately but the manual part of such operations needed some scientific means to estimate accurately the time required to perform them. Two other great industrial scientists, Frank B. Gilbreth and Dr. Lillian M. Gilbreth, introduced and pioneered a philosophy of thinking about manual operations and an orderly arrangement for precise measurement of them. Although their objectives were economic and humanitarian in that they sought the one best way of performing operations and the elimination of industrial fatigue, they created the structure upon which time values for fundamental motions were studied and built.

The past decade brought to completion the development of essential fundamental motion-time values. It also brought widespread application of them as a means of prediction. Production planning has been a chief beneficiary since industrial operations can be completely planned before they are started. They can, moreover, be replanned if the economics prove to be unfavorable. Apart from achieving best methods, operations in progressive production arrangements can be balanced before they are actually begun. In essence, scientifically developed fundamental time values have given industry a vitally important means of prediction enabling it to keep its production planning separated from and adequately in advance of its production performance.

Production-planning executives of this and future decades owe much to Harold B. Maynard, G. J. Stegemerten, and John L. Schwab for the painstaking researches that resulted in these values, techniques so essential to scientific prediction, and their generosity in releasing their findings for broad application in industry.

Stabilizing Employment

In the control of production during the past decade two disciplines have been refined and extended. One of them seeks the achievement of control by economic regulation of inventories. Based upon a company's schedule of net working capital and its consideration of customer service, a policy is established which states that stocks of finished-goods inventories shall not exceed a certain number of weeks of current sales for each item produced and also shall be not less than another certain number of weeks. When a company's sales are seasonal, inventory upper-stock limits are reached and production is curtailed or temporarily discontinued. This, of course, results in employee dislocations, re-

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training, bumping, poor community relations, and many hidden costs such as those for increased unemployment-insurance taxes.

The other discipline of production control seeks to level production operations and stabilize employment. Based on accurate sales forecasts production is spread over the entire year at an almost level rate. This creates excellent employment and community relations. It is claimed by companies employing such controls that the effectiveness of operations is greatly increased when fears of cut-backs are removed. The primary requirement to achieve such planning and control is working capital to carry inventories produced during seasons when sales are low. There are also the risks of possible falling prices to be considered.

Companies struggling for financial survival will of necessity have to control inventories within limits because they have not and cannot obtain sufficient working capital to do otherwise. With present high corporate taxes and the increased cost of doing business because of inflated values it becomes increasingly problematical as to whether those companies who have so courageously achieved level production will have enough money left after taxes to operate under so admirable a plan.

Government Regulation

During the period from 1932 to 1942, industry was reasonably free from government regulation. Production forecasting, planning, and control were relatively simple and needed only the techniques that were then available. The past decade brought a hot war and a cold one that continues. Both of them have brought government regulations with dislocations and confusion for the production planner.

The Controlled Materials Plan regulated production on a national scale but its enormity made it sluggish. Production planners and purchasing executives spent almost as much time completing government forms as they did in performing their essential company functions. Price regulations produced equally dislocating results. Many substitutions of raw materials were caused by price regulations based on artificial factors that established prices for commodities out of traditional relationship with each other. These regulations stimulated the application of all known available production-planning and control devices and the development of several new ones. They also proved American industry to be a virile, ingenious body of producers capable of working under many handicaps and artificial influences. Early in the next decade the planning of production should be simplified by a return to natural operations freed of most government regulations. If regulations continue, industrial planners have the skills to cope with them. No technical problems arising in their work can cause them to fail.

Standardization and Simplification

One important factor that has aided production planning and control during the past decade has been an increasing awareness of the benefits to be obtained by standardization and simplification. As a result of the educational effort by the American Standards Association, Inc., many companies have organized standards committees. Product standardization and simplification have eliminated the planning and control of many items requiring expensive setups and unprofitable operation. Standardization of component parts used on several products also has been advantageous. Standardization of raw materials has reduced the number of items carried as raw stock and thereby reduced working-capital requirements.

The excellent work of the American Standards Association continues to urge and implement the adoption of standards embracing almost every industrial and scientific activity. The United States Bureau of Standards also has continued to make important contributions in providing increasing accuracy of measurement, opening new applications of standards and refinements in old ones. Simplified Practice Recommendations introduced by Herbert Hoover when he was Secretary of Commerce, have been continued and extended by common agreements within industry and commerce. There is a tendency for other governmental agencies, which are less qualified, to set standards when voluntary co-operative agreements by members of industry fail. For this reason, in the coming decade, many companies and industries will use the facilities of the American Standards Association to resolve differences and to establish many new industrial standards and simplified practices.

Research in Organization

Since the last report industry has conducted many research projects in the field of organization. Success with physical decentralization of operations has led to the development of decentralized organization where physical decentralization itself has not been justified. There has been a real awakening to the advantages to be obtained by breaking large complex problems and organizations down into several smaller ones that are manageable.

There is evidence that centralized sales forecasting is effective especially where there is sales interaction among products. Many centralized production-planning and control organizations, however, have been divided into smaller separate organizations. This division in some cases has been by product lines and in other cases by class of work. There also are cases where divisions have been made by types of raw materials used. The principal reason for the effectiveness of these organizational decentralizations is that fewer people can understand and deal with all of the factors of production planning and control in their smaller respective spheres of activity. Another reason is that the production-planning and control functions can be made a part of a closely integrated organization reporting to a unit manager who can operate with almost complete autonomy, cutting through the traditional red tape characteristic of large centralized organization.

Inception of Statistical Quality Control

Production planners always had two unpredictables with which to contend. Even though purchase requisitions for the correct raw materials to be delivered at the right time had been placed, there was always some chance that some unpredictable portion of the raw material would be unsuitable for production when it arrived. Production schedules also may have provided correctly the needed quantities of finished product but some unpredictable portion of production was lost because it failed to meet quality standards.

Until shortly before World War II, these two factors gravely reduced the accuracy of production planning. Then, under the stimulating leadership of Dr. Walter A. Shewhart, and because of the demand for increased production, industry rapidly adopted statistical quality control as an instrument for reducing scrap and assuring more uniformly acceptable quality. The advantages to production planners came as auxiliary benefits.

Today, by using statistically sound sampling plans, companies can determine quickly whether raw materials conform with specification requirements before production operations have to be delayed or discontinued and nonconforming materials replaced. Records of quality ratings of each supplier's raw materials indicate the nonconformance probabilities of each. Adjustments can therefore be made in ordering lead-times in cases where rejection of some shipments is to be expected. More important is the advantage of giving technical assistance to suppliers whose performance proves to be unacceptable. As acceptance sampling experience and skills are extended relaxed, normal, and increased sample sizes are used to achieve a high degree of confidence that

nonconforming materials will be rejected and conforming materials accepted. This feature not only permits the most economical inspection of raw materials but it also provides a high degree of confidence in production planning.

In the past, production planners had no accurate basis for estimating how much production would be lost at each operation because it failed to meet specification requirements. Today, quality-control engineers determine process capabilities of equipment and processes. These tell the production executive the percentages of product that normally can be expected to be lost all along the production processes. Allowance for these losses can be included in the production schedules as an expedient means of increasing planning accuracy. But richer opportunities for improved and stabilized production output are generally obvious from the quality-control data. Production executives soon learn that certain equipment is inherently incapable of producing within the desired quality limits. Modifications or replacements are made. They also discover that certain operators produce less nonconforming product than other operators. Selection and training of operators removes this source of loss and variation. Some raw materials are found to produce less scrap than others. Specifications are modified to require those materials that give best results.

All of these quality-control factors, brought into play during the past decade, have helped to improve the determinative work of production-planning executives. As planning becomes more accurate, control of production is simplified.

Ratio-Delay Analyses

When production executives awakened to the great opportunities offered by statistical quality control, they reasoned that small samples could serve also to evaluate the effectiveness of operations. During the latter part of the past decade a new statistical technique called "ratio-delay analyses" was developed and applied to industrial operations. Industrial engineers traditionally have studied carefully the time necessary to perform the elements of operations. When these operations were combined in a complex and lengthy process, final outputs were generally considerably less than expected. Detailed analysis of what was happening in the whole process was difficult if not impossible because many of the processes covered large areas, parts of which were beyond the vision of one observer.

Evaluation of process effectiveness by a ratio-delay analysis is simple and workably accurate. It is based on small samples to provide a statistical inference of operating effectiveness just as small samples tell of quality performance as production continues. An observer simply walks through the area in which a process is operating for a short period of time (say, 10 min) and observes delays and the causes for them. These short-period observations are repeated at random intervals until the cumulative average ratio of delays becomes constant or sufficient observations have been made to establish statistical control limits. Observations are classified as avoidable delays, unavoidable delays, and productive time.

Other more detailed classifications also can be used. When the analysis has been completed the production executive can state with confidence that his process effectiveness is a certain ratio or percentage of what can be expected as ideal. Moreover, he can point to delays that are unavoidable; those that are avoidable and can be eliminated. Modifications of equipment and preventive maintenance also can reduce or eliminate those delays that originally have been classified as unavoidable. This technique can be used to evaluate and improve operations. It also can be used by the production executive to predict his operating effectiveness before the day is ended by an examination of small statistical samples taken during the day.

Operations Research

World War II presented many military problems requiring choice of weapons, methods, and approach. A new applied science known as operations research was born of the necessity for making good decisions supported by sound quantitative bases. This new science proved to be so successful in military operations that industry recently has been attracted to it for the solution of its production forecasting, planning, and control problems. It transcends the limitations of economic production lot-size determinations and combines with linear programming approaches to cover all the ramifications of consumer wants, distribution, production, and sales. When all factors possible of quantification have been so studied it should be possible to make management decisions with greater confidence. The application of this science to problems of production management is too new for any overwhelming testimony of its effectiveness to be given.

Inventory-Pricing Methods

Methods for pricing inventories have been given serious consideration during the past decade. Many companies have modified their inventory-pricing methods. This has required the adoption of new inventory-control procedures.

In previous decades when fluctuations in commodity prices were not wide and inventory turnovers were high, many companies enjoyed the advantages of the standard-cost method of cost accounting. Costs were established by revising periodically the costs of materials according to prices which were expected to be quoted for delivery during a specified period. Revisions were not made at rigidly specified times but kept pace with changes in the market. Under this arrangement costs were determined with reasonable accuracy even before stock withdrawals and production. Adjustments for the difference between invoiced cost and standard cost were determined and charged to purchase variance.

The "average-unit-cost method" was and continues to be used in job-order production, where customers' orders must be priced equitably on the basis of the average cost of material used in filling each order.

The "first-in-first-out method" is used where raw materials account for a relatively small part of the production cost and where manufacturing costs must reflect the actual sequence of price fluctuation for raw materials.

Government regulation and price controls during the past decade have brought the "last-in-first-out method" of pricing inventories into prominence. Price fluctuations during this period have been so violent that an inventory-pricing method permitting stock to be carried on the books at old prices, so that inventory values do not change greatly from one accounting period to the next, was needed. Current fluctuations in prices of raw materials had to be more realistically reflected in the current cost of sales. "Lifo," as this method is called, meets these requirements and has been adopted by many companies in which the raw-material component is the major factor in total cost. It also equalizes profits to a considerable degree by bolstering them when prices are falling and limiting them when prices rise. Under present tax structure this equalization of profits is essential because high profits in one tax period cannot be retained to offset lower profits or losses in another.

Inventory Turnover

All of the factors limiting the availability of working capital and increasing the cost of doing business have exerted great pressure on production and financial executives to find ways and means of increasing inventory turnover. These adversities created a new type of thought and action with respect to inventories. Typical business enterprises are not fully continuous nor

are they intermittent. They are, rather, a combination of several different types of production, the values of the items of which vary widely. Analysis of inventory discloses the approximate 15 per cent of the sales items that comprise the approximate 85 per cent of the total inventory value. Then shortened process intervals, special handling, ordering, and expediting of these items are developed. This new concept of control has brought about some astonishing increases in inventory turnover.

Throughout the years industrial and commercial operations have been emitting signals to management. Management is slowly learning the language of these signals and how to respond to them. It has learned how to make some proper responses to signals made by sales and to adjust its sales forecasts and production schedules. It has become sensitive to the signals and warnings sounded by production processes and interpreted by quality-control and ratio-delay-analysis techniques. Industrial and commercial indexes are being refined to the extent that many of them foretell the trend of events to come. Cybernetics is attempting to find the common elements in the functioning of automatic machines, business systems, and the human nervous system, and to develop a theory which will cover the entire field of control and communication in machines, business organizations, and living organisms. Many of the responses to signals operate in the same way as do servomechanisms maintaining constant adjustment.

Promise of the Future

The implications of the future are exciting and stimulating. Will the next decade bring the automatic factory, with automatic control of production and operations? Will it bring capabilities for interpreting human attitudes, wants, and needs so that sales forecasting can be perfect? The answers to these questions present a predictive risk but it is safe to say that the progress made during the past decades will seem pitifully small in relation to the progress that will be made in the decades to come. Ten years hence management skills will contribute even more importantly to advancing human welfare.

Government will awaken to the fact that management can control better its industrial and commercial activities than can federal agencies. Government also will learn to apply management skills to its activities thereby improving its effectiveness and reducing the burden of abnormally high taxes and deficits. These are primary because they will recreate the industrial ability to finance production improvements and expansions and permit the working-capital reserves that are needed to insure stable production.

Predictive skills will improve and better management decision will be made through the broader application of sales forecasting, quality control, fundamental time values, ratio-delay analysis, and operations research. Many advances in production planning and control will come from modifications of organizational arrangements. Pricing of inventories will receive considerable attention and many companies will adopt the Lifo method. Inventories will be maintained at low levels until tax relief permits retention of more working capital to finance them. All contrivances will be employed to increase inventory turnover.

Extensions of and improvements in production forecasting, planning, and control in the next decade will have far-reaching effects in the management, social, and economic realms. They will help to strengthen our system of competitive enterprise and make it serve better the interests of all segments of our social structure.

Work Simplification and Work Measurement

By DAVID B. PORTER[1] and ERCOLE ROSA, JR.[2]

In this paper the term "work simplification" is considered to mean the application of motion, process, and procedure study techniques for cost reduction through appropriate training at all levels of supervision and management, and even below the supervisory level. The term was applied by Mogensen in the mid-1930's to the Lake Placid Conferences and later by those who were using the training approach to methods improvement to distinguish this from the older system wherein methods improvement was the sole function of the motion-study specialist or methods engineer. This newer approach has proved to be the step which greatly overcomes the barriers of resistance to change on the part of line supervision and workers, and has extended the usefulness of the methods engineer in supplying asked-for technical assistance to line supervision.

Early Training Programs

The early training programs were directed mostly to foremen, supervisors, and methods men. The training was done mostly by the lecture method and instruction given in the use of the process charts, with the expectation of their adoption and use by all supervisory people taking the course. It was discovered that this was not always true and that it took a great deal more to win the acceptance of the use of these tools. It was recognized that it took practice in the use of these techniques in order to give confidence in their effectiveness.

In the earlier days when the training emphasis was placed more on the individual, after a training program was finished and the men returned to their respective departments it was sometimes difficult to keep up the interest of the individual who might feel somewhat alone, and lack confidence at first in his ability to use the techniques.

In order to overcome this it was found advisable to break the training groups into no more than 10 or 12 and to present the material by the consultive and participation method, rather than by the older lecture approach. Out of this there grew a new realization that a group approach to the solution of methods problems is far more effective than the single approach. The reasons for this are twofold: (1) A better solution is found when more minds are directed to the problem; (2) the group approach permits all those in any way connected with the problem to have a part in its solution, which thereupon makes the adoption of the improved method immediately acceptable to all concerned.

The simplest change in an operation will affect no less then two people, the foreman and the worker, and usually the methods engineer as a third person. If the problem is at all complex it may involve a layout engineer, safety engineer, tool engineer, and higher echelons of plant supervision.

After the completion of the training program this participation approach is now being applied successfully to the continuation of work simplification through the forming of project groups or com-

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mittees meeting regularly. These groups are usually made up by departments and may meet to solve a particular problem or project, or they may meet regularly, say, every 2 weeks, to carry along an agenda of various projects upon which they are reporting continual progress. This type of approach brings in people farther down the line to participate in the solution of these problems. Companies such as Wolverine Tube Division of Calumet & Hecla Company, and Standard Register Company, H. P. Hood & Sons, are making effective use of group participation.

As an aid to the training program and to the continuing work of groups, a laboratory or workshop adjacent to the work-simplification office, where foremen and supervisors can come in and work out their own ideas for improvement, has been found most effective. The opportunity to try out the principles of motion economy with practical examples in a laboratory is a most effective way to teach these principles and give foremen and supervisors confidence in their use and application. Many schools and universities teaching motion study and the principles of motion economy place considerable importance on the laboratory phase of the work.

Whereas the earlier training was directed mostly toward foremen and people on the lower supervisory level, it was later realized that higher levels of management, including top management, must have an understanding of these techniques in order to give proper support to their use, and especially to recognize management's responsibility in the social problems involved through technological changes. Therefore work-simplification conferences for executives have been developed, and in most plants using work simplification, appreciation sessions are held for top management.

Inception of Work-Simplification Idea

Although work simplification actually began about 20 years ago, it could be said that it has had its most rapid growth in the last 10 years, partly due to the stimulation of World War II. Some of the war industries, particularly the aircraft industry, entered upon large-scale work-simplification programs to train rapidly many thousands of supervisors. The Bureau of the Budget in Washington, and the Army in the Services of Supply and Signal Corps in the Pacific and Mediterranean theaters carried on active programs, and since the war the Army has been carrying on, up to the present time, programs in the Army of Occupation in Germany.

The fields of work simplification have not been confined to industrial application alone, but have branched into commerce and business, and many others, such as libraries, hospitals, and the home.

The department stores within the last 5 or 6 years have been taking a very active interest, and within this period a rapid growth has developed in the clerical field among insurance companies, utilities, banks, and in the offices of many large industries.

On an industry basis work simplification has been carried to a number of European countries, such as England, France, Sweden, Norway, and Germany, where they have either sent men here to attend work-simplification conferences, or American engineers have conducted programs abroad.

The Future of Work Simplification

In addition to an expansion of the trends already noted, it is fair to predict a much further extension of work simplification to all levels of management, from the very top down to the last person in the company. Development of some of the new techniques, such as sound on magnetic tape, and the new magnetic sound film, should do a great deal to promote interest and enthusiasm in the before-and-after films; greater use of the newer developments in visual aids of all types; further use of memo motion, which is the taking of pictures at intervals of 1 sec, or longer, in order to study the work of related groups of people; and the use of the chronocyclograph technique, which was developed by the Gilbreths and which has been used effectively by the Anne Shaw organization in England, and by Harold Dunlap of H. P. Hood & Sons. This is a most effective technique in training the operator, and in selling the improved method.

Work Measurement

The increased attention paid to the techniques of work simplification during the decade ending in 1952 was paralleled by a similar increase in the discussion and application of the techniques of time study or work measurement. The term "work measurement" has come into more prominent use because of the trend toward considering the preparation of a production standard as the application of a wide range of knowledge including the range of human capacities, the implications of the science of measurement and the application of scientific methods of analysis.

The demands for increased production which continued throughout the past 10 years resulted in a great expansion of the application of production standards. Many workers not previously covered by systems of work measurement were brought under its influence. Training in the application of work-measurement procedures has become an important activity in industry, in government, and in the university.

There also has been a significant increase in the amount of critical analysis devoted to the techniques of work measurement. The increase in this critical attention was due to several factors.

The first of these factors is the pressure on management to continually improve its operating procedures in the face of competition and other business influences. The rising cost of direct labor has proved to be an extremely important influence. The need for more accurate and more effective controls upon all phases of operation and which depend upon reliable work measurement has had its effect.

The growth in the relative strength of the unions has exerted a considerable pressure toward the improvement of the procedures of work measurement. Large and capable union research staffs devoted their efforts to determining the relation of work measurement to the collective-bargaining procedure. The results of this study have tended to emphasize the part played by human judgment in the preparation of production standards. The union demand has been for procedures whose accuracy would fall within the range of "the percentage increment or decrement to the wage scale involved in collective bargaining negotiations" (1).[3]

The increased recognition of the importance of work-measurement techniques as a phase of the engineering curriculum has resulted in the training of larger numbers of science-minded individuals who are devoting themselves to the improvement of the existing procedures.

The attention given to the techniques by workers, such as Ghiselli and Brown (2), Ryan (3), among others, in other fields of science has tended to increase the rapidity with which significant thinking has developed with regard to work-measurement procedures.

The combined efforts of these groups have produced several important results. These include the improvement of existing techniques, the development of a more scientific foundation for the evaluation of work-measurement procedures, the utilization of methods of analysis originated in related fields of science, and the application of work measurement with useful results in many situations far removed from the job shop and the production shop.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

The Critical Approach. Critical comments concerning the procedures of work measurement have appeared at various times since the publication of the first paper by Taylor (4) on elemental time study in 1895. Early union criticism resulted in the introduction of a rider into appropriation bills passed by Congress seriously restricting the application of work measurement in government departments. Hoxie's critical comments stated many years ago are still repeated with considerable justification. The critical attitude expressed by the Gilbreths did much to create a more realistic attitude toward the implication of scientific procedures applied to the preparation of production standards.

In the decade ending 1942, several papers critical of the procedures of work measurement written by researchers active in the fields of psychology and physiology were published (5, 6). Reacting to these papers and those that followed, Presgrave wrote in the preface of his book, "Dynamics of Time Study," . . . . "In the last few years we have had a plethora of books in which somebody 'looks' at something. Usually the 'looker' is also an outsider. Poets look at engineers, engineers look at love, and so ôn, but rarely does anyone look at his own profession. Had the cliché not been overdone, this book might have been called 'A Time Study Man Looks at Time Study' " (7).

To Ralph Presgrave must go the credit for the first serious examination of the basic concepts which form the foundation for the techniques of work measurement prepared by a time-study practitioner. The flow of critical material increased in intensity with the stimulation provided by Presgrave and reached a peak with the publication of Gomberg's (1) most valuable analysis of work-measurement techniques.

Work Measurement and the Present

With this background of the major trends in the past decade, a summary will be presented of the specific developments as they resulted from the dynamic interplay of the various forces at work in the industrial situation. These developments will be examined under three headings:

(*a*) Basic concepts of work measurement.
(*b*) Current practices.
(*c*) Extent of current application.

The paper will conclude with a brief examination of the outlook for the next 10 years and a statement of the methods and developments likely to have an important bearing upon the application of work measurement in industry.

Basic Concepts of Work Measurement. Production standards are important to management because they provide the basis for managerial action in many areas. Effective production standards contribute to production planning, estimating, plant layout, and a host of other management controls. In the basic cycle of executive action involving specification, performance, and review, the knowledge of the time required to perform a specified task must be combined with the specification for the successful performance of the task to produce a basis for the evaluation of the final performance.

In this sense the production standard represents a prediction and at the same time must satisfy all the requirements of objective measurement. The production standard involves a prediction that the conditions which today permit the performance of the specified task in a determined length of time will continue to exist in the future so that if the specified task is performed in the specified manner it will take the same amount of time to perform. As a measurement system it is necessary that a suitable definition of the quantity to be measured be prepared, and that "a true, or perfect, or correct, process of measuring the quantity be available" (8).

The concept of stability, as developed by Shewart (9) and applied by him to the "economic control of quality of manufactured product," and applied to industrial production rates, as suggested by Gomberg[4] and demonstrated by Abruzzi (10), has been shown to be a valuable tool available to the engineer for the detection of work environments in which successful prediction may be attempted. Rigid adherence to the levels indicated by the control chart, however, neglects the intentional influence that may be exerted by the human factor in the work situation and also fails to take into account the ability of the engineer to determine by observation if a satisfactory level of production is being maintained.

The implications of measurement as applied to time study have been examined in great detail by Presgrave who makes two important observations: "The first is that as time advances measurement takes on new accuracies. The second is that it takes on new forms."[5] This second observation leads to another—"Since measurement is essential to control, and since control of costs is essential to competitive enterprise, it is small wonder that labor should come under the rule of measurement, for labor is the main element in cost, perhaps the sole element in the last analysis."[6]

Continuing his analysis of time study and measurement, Presgrave concludes, "time study, *per se*, is concerned fundamentally with two measurements and two measurements alone. One of these is the measurement of elapsed time, i.e., the true basic measurement. The other is the measurement of the speed of the operator's motions, i.e., the measurement of the correction factor. There can be no other basic measurement and no other correction factor within the concept of time study as measurement"[7]

Realizing the magnitude of the problems that are left untouched by this approach, Presgrave admits that "some compromise between the theoretical and the practicable is inevitable."[8]

It is this last conclusion which summarizes the most prevalent current attitude of the serious practitioners in work measurement.

Current Practices. The area of work measurement that received a great amount of attention during this decade was the phase of the procedure criticized most often as introducing a high degree of judgment into the preparation of production standards—"rating." Progress toward the development of improved rating procedures took two forms. The first was the development of national benchmark-rating films showing known levels of production output. The other was the development of new systems of rating.

Benchmark-rating films received considerable attention as the result of the need for films that could be used to increase the consistency in rating within company organizations and throughout the country. The two important projects were conducted by Prof. Ralph Barnes (11) and by the Society for the Advancement of Management (12).

In the SAM project, films of typical industrial operations were shown to more than 1200 trained time-study men and industrial engineers in all parts of the country. The combined ratings of these men were used to produce 24 film sequences presenting common industrial operations at known levels of production. The completed films have been made available for use in industrial and university training in rating.

The SAM rating project produced significant information concerning the rating procedure.[9] It was found that there was a high level of consistency among all the observers concerning the final production standard although there was no identifiable single concept of "normal." It followed from this that there was a close relationship between the rating factor and the allowance factor,

[4] Reference (1), p. 37–39.
[5] Reference (7), p. 19.
[6] Ibid. p. 26.
[7] Ibid. p. 31.
[8] Ibid. p. 32.
[9] Reference (12), pp. 105–107.

where low rating factors were accompanied by high allowance ratios and vice versa. Useful conclusions concerning the types of errors made in rating also were drawn from the data.

In the past decade procedures for determining the number of observations required to produce specified levels of reliability were developed. Using procedures originally applied in statistical quality control Mundel (13) suggested a method for determining the number of observations based on the use of the sample standard deviation. A somewhat different method was developed by Wilkinson (14), who used the ratio of the square of the sum of the observations to the sum of the squares of the individual observations.

Although the interruption study continued to be used widely for determining the incidence of delays in production, the ratio-delay method, based upon the statistical properties of the binomial distribution, was accepted rapidly for this function because of its many advantages. This method developed by Tippett (15), and given great prominence by Morrow (16), prescribes the taking of a large number of observations of the work situation in a random manner. The distribution of delay occurrences in the sample gives a close approximation to the frequency of occurrence of these delays in a long period of time. The application of the control chart in conjunction with the ratio-delay method by Abruzzi (17) produced an extremely useful tool for the analysis of delays.

The further development of procedures for the collection and application of standard data was emphasized to reduce the degree of judgment required in the preparation of production standards. The research effort was directed toward more precise definition of element variables and of motion elements and toward greater precision in the measurement of motion-time values. In part, this research was concerned with answering the criticisms leveled against the basic assumptions of the standard-data procedure by Gomberg[10] and others.

Significant progress toward the development of universal micromotion standard data occurred with the introduction of the MTM (18) and Work-Factor (19) systems. The willingness of the originators of these newer systems of standard data to make their original data available for general analysis gave promise of the early development of a completely satisfactory system of micromotion standard data.

The accurate specification of the method to be used to perform the task is essential to effective production standards. Because of this, methods for recording the work method were investigated. The memomotion method for filming operation details was developed by Mundel.[11] It produced several advantages over the standard method of filing operations with the 16-mm camera. The chronocyclegraph technique developed to a high state of utility by the Gilbreths received relatively little attention during this decade although much useful work was done with it in England by Anne Shaw (20).

Extent of Current Application. In the past decade production standards were applied more intensively to operations which had not been covered previously. The number of employees performing clerical operations made this an extremely fertile area for the development of clerical standards (21). Maintenance and other indirect labor operations were covered more adequately by production standards. The benefits of extending the coverage to indirect workers was well recognized and emphasized by many engineers, notably Carroll (22).

The Future of Work Measurement

Research devoted to the development of improved work-measurement procedures will take many forms, will be concerned with every phase of work measurement, and will involve many areas of science. The complex nature of the problems encountered in work measurement will put a premium upon the use of multidisciplinary research teams, as suggested by Korn and Prian (23) where engineers, psychologists, physiologists, and others work together. The mathematical and statistical procedures of "operations research" which produced such outstanding results in the solution of military problems, and which also involves the use of combined teams, will find important application in work measurement.

Some of the research in the area of "cybernetics" should prove useful to the handling of problems of work measurement. Some indication of the possible value of this information is provided by the data presented by Mayne (24).

Further development of benchmark-rating films and the wider use of these films will continue with a consequent improvement in the general level of rating accuracy. The relative importance of rating may decline, however, as the current research concerned with standard-data procedures continues to produce improved techniques.

Present knowledge concerning the capacity of human production is relatively limited, but will increase. The wide range of production output that exists from plant to plant, and even within plants, suggests that there are influences in the work situation which are barely understood.

Conclusion

The 10-year period ending in 1952 produced many important developments in the theory and practice of work measurement. These resulted in a fuller understanding of the scientific basis of work measurement, in specific improvements in the techniques, and more widespread application of work measurement in industry.

[10] Reference (1), pp. 56–165.

[11] Reference (13), pp. 295–298.

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Wage Incentives

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General Comment

A SURVEY in *Fortune* (1)[2] finds that only about one half the production employees are really interested in their work. *Modern Industry* finds 60 per cent. Anyway you look at it, this is a problem that can be solved perhaps better by selection and placement than by any type of incentive.

We might find some differing percentages if groups making the surveys had distinguished between people working on incentive and those on daywork.

The latest survey (2) available reports 34 per cent of selected manufacturing plants have incentive plans with 30 per cent of the employees covered! Two thirds of the employees and 85 per cent of apparel establishments were on incentive. Roughly 40 per cent of the employees and 70 per cent of textile plants utilized incentive plans. Twenty-five per cent of employees and 17 per cent of metalworking plants had incentives. Chemical plants were low with 7 per cent of their employees and 6 per cent of their plants. Important industries not included are steel, printing, rubber, and lumber.

Interesting is the extent of incentive applications in nonmanufacturing. Briefly, the percentages of employees covered were 37 in automobile repair shops, 22 in bituminous-coal mining underground, 34 in clothing stores, 28 in department stores, and 14 in power laundries.

Of course, where there is little or no interest, wage incentives probably can be no more than a remedy.

However, experiences seem to show that certain interests are created by incentives. Making more money gives many people the feeling of "getting ahead." Learning one's relative skill status through comparison of output percentages is another form of success measurement.

Wartime Incentive Plans

During World War II, wage stabilization was set up to keep the lid on inflation. Wage increases were limited. The "Little Steel" formula was invented to judge applications for increases. Regulations covered incentives.

To install or extend an incentive plan, approval by the Board was required. The basic limitation was that unit costs should not rise. Reports were requested that would show what did happen to earnings and productivity after incentives were applied. Here it is interesting to note that plans to pay for increases in output above past performances were limited to payments on a 50-50 basis.

John W. Nickerson laid out sixteen "guiding principles" (3) to assist those approving incentive plans and those starting them. Despite these, roughly 70 per cent of the plans introduced were based on past performances. "The history of wage incentives shows that it is this very basis that led to practices and conditions which placed sound wage incentives in a bad light and led to their failure and discredit" (4). We can say, "This was an emergency," but we haven't made much progress when we repeat past mistakes to that extent.

About 15 per cent of the plans started were coverage plans. They were plantwide incentives adopted to pay everybody for getting out more war production. These had the desirable feature of paying incentive to indirect people. Group incentives made up 41 per cent of the wartime plans discussed before this Society in December, 1943.

Their chief fault was the inability to adjust correctly for the multitude of changes in method, design, and production.

Despite the wage-stabilization controls, many incentive plans were allowed to become badly distorted. Standards were set loosely on purpose. Method improvements were overlooked in some plants to allow take-home to rise. Top limits on output were taken off. Some irregularities were injected to raise earning. Some existing incentive plans went haywire.

Part of these defects dropped out when war production ceased. But many problems were carried over because many of the same employees shifted to producing civilian goods. After the war, bargaining, strikes, and re-engineering were used to correct some of these errors. Many of the variations are still with us (5).

Incentive Ratios Decreased

In this period, many concerns granted wage raises that were paid as cents per hour for the hours worked. Some repeated the practice for several rounds. This caused increasing proportions of incentive-sharing plans. Suppose 80 cents was the initial base rate and 20 cents the premium earning at 25 per cent above the standard. When a series of raises brought the wage to $1.20, the premium should have been 30 cents for like performance. But, in our example, it stayed 20 cents after these general increases. Hence the incentive factor dropped to 16.7 per cent. Probably this happened most often with piecework plans. But there were many instances in plants using standard time. This took place while there was insistence upon changing sharing plans so as to pay one-for-one.

Incentives for Indirect

Incentives for indirect work of all kinds have been largely neglected. Barnes shows that only about 20 per cent is on incentive (6). This has brought on demands for wage grants to offset the "lost opportunities." Such arbitrary increases not only fail

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[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

to provide incentives to increase productivity, but also they reduce the actual incentive for those who are on some plan of measured work.

Some progress in overcoming this deficiency is evident. Office work is being studied (7) and incentives have been set for maintenance work (8).

More attention is being given to the study of distribution costs (9) and to the setting of more objective standards for the rating of salesmen (10). Incentives are paid for increasing the profitableness of sales efforts (11).

Incentives and Collective Bargaining

Some unions have set up staff departments to work on time study, wage incentives, and like technical matters. Some have issued pocket handbooks on these subjects. Staff men advise local officers and stewards on techniques.

Some unions oppose incentives of all kinds including profit sharing. Others reflect attitudes ranging from co-operation to toleration. Among these are degrees of interest extending from insisting upon codetermination to limiting their concern to the use of grievance procedures. Grievances may be complicated by political issues and campaigns to develop their number.

Union contracts vary greatly. Some have clauses that pertain to time study and incentives. Bearing on this is the government regulation that requires union consent to the introduction of wage incentives. In contrast, ". . . in a great many companies detailed time study plus an active wage-incentive plan coexist with a rigorous union and a forthright labor contract which makes absolutely no mention of time study" (12).

One clause that suggests progress in developing productivity is used to define a "fair day's work." The definition is, "a fair day's work is that amount of work that can be produced by a qualified employee when working at a normal pace and effectively utilizing his time where work is not restricted by process limitations. Example: A normal pace is equivalent to a man walking, without load, on smooth, level ground at a rate of three (3) miles per hour" (13).

Another agreement suggesting progress is the oft-reported General Motors' contract with UAW-CIO of May, 1950. "An annual improvement factor of 4 cents per hour to be added to the base rate starting May 29, 1950, and 4 cents additional annually for the life (5 years) of the Agreement." . . . "The annual improvement factor provided herein recognizes that a continuing improvement in the standard of living of employees depends upon technological progress, better tools, methods, processes and equipment, and a co-operative attitude on the part of all parties in such progress. It further recognizes that to produce more with the same amount of human effort is a sound economic and social objective" (14).

"Toward that end the union agreed not to allow any 'restriction of production or interference with production of the Corporation' (117). New machinery, technological improvements and new production methods can be introduced without union objection. In recognition of the need for maintaining productivity, the union gives management full authority to set production standards, subject to the grievance procedure (77-78). Wage payment plans can be installed (90-91)" (15).

Measured Day Work

All the while, a sprinkling of plants has changed to or adopted measured day work. This is an incentive plan when extra money is paid for extra production above standard. One method is to adjust the base rate each 3 months according to the average performance for the past 3 months. Such a plan has the advantage of stabilizing earnings. It might be called measured merit rating. The plan works with improving performances. However, in some companies, base rates did not go down when productivity fell off. Sometimes performances fell to preplan averages while payments stayed at the premium level.

Suggestion Awards

Progress in the field of employee-suggestion plans is being fostered by the National Association of Suggestion Systems. Surveys it conducts indicate an average of roughly 200 suggestions per 1000 employees with about 25 per cent adoptions paying $21 average awards. Projecting from these surveys, it is estimated that the potential incentive is $30,320,892 annually for those eligible (16). Noteworthy here is the annual rate of change in productivity of 15.35 per cent in Lincoln Electric. Another potential can be seen by comparing progress of this company with 3.11 per cent in all manufacturing and 2.74 per cent in the machinery industry (17).

Besides the obvious values, "suggestion systems are effective as a method of improving industrial relations. . .For the employee, a suggestion system forms a creative outlet. It makes him feel that he is a part of the whole and is contributing to the growth and progress of his company" (18). Sixty-nine per cent of the respondents in the NASS 1949 survey "felt that improved industrial relations was the greatest benefit, with monetary benefits second."

Beyond these, we should remember that "the great rise in real wages in the past and the gains in productivity that made it possible, have not come about automatically. And future advances will not be rapid unless we have many important scientific discoveries. . ." Earlier this report says, "Since there are so many places where an improvement may originate, enterprises must be on the alert to improve their methods and products" (19).

For continuing progress, more incentives can help. "The incentive to improve is of greater importance, and yet is seemingly less prevalent than the ability to invent when some specific objective is to be attained." (20). Perhaps a solution lies in the plan proposing (21) that suggestion awards be paid out in parts to the suggestor, to others directly affected, to the department pool, and to the plant pool.

Profit-Sharing Plans

Since our last report there has been a rising interest in and use of "profit-sharing" plans. "Trusteed plans for profit sharing have increased from about 300 in 1942, to over 4000 in 1950. Membership in the Council of Profit Sharing Industries (founded June 13, 1947) indicates that there are twice as many current plans as trusteed, all registered and qualified with the government" (22).

The plans are many and varied. *Factory Management and Maintenance* reviews eight types (23). One source classifies and describes them in three groups as follows:

1 Cash and wage-dividend plans such as used in Eastman Kodak Company, Whiting Corporation, and Pitney Bowes, Inc.

2 Production and cost-saving plans like the Scanlon Plan, Rucker Share of Production plan, and Lincoln Electric Company's plan.

3 Other profit-sharing possibilities similar to the plans in Daisy Manufacturing Co. and Willey-Wray Electric Co.

A Conference Board survey divides 167 responding replies into two groups with 67 current and 100 deferred distribution plans. "The deferred distribution plans have been introduced largely as a substitute for a straight pension plan. . ." (25):

1 Eighty-five per cent are in manufacturing plants.

2 Thirty-two per cent are in machinery and metal products.

3 Eighty-seven per cent are plants having less than 1000 employees.

Probably, the many different reasons for adopting "profit-sharing" plans cause the several varieties of designs. Basically, however, most seem to strive for creating better teamwork. They try to bring in the relations between the extra earnings and the success of the company in providing job security. Obviously, this ties in with wage adjustments and tends to reduce excessive demands.

Union attitude toward profit sharing is mixed, as we may conclude from two statements made in the Conference Board survey:

"Union opposition has been one of the major causes for discontinuing profit-sharing plans, . . .The attitude of union employees is found to be entirely satisfactory in a third of the companies with deferred plans and fifteen per cent of the companies with current-distribution plans."

In this connection, available information emphasizes the need for wage scales that equal or exceed the community rates for similar work. Also, the point is made that an important part of the plan's success depends upon informing employees about the business principles involved as well as keeping them posted on current conditions. Finally, it appears that better results are shown in the smaller plants where the individual more nearly sees the effects of his own efforts on the profitableness of the company.

For profit sharing to be an incentive plan seems to depend upon the latter two elements; (*a*) understanding of competitive enterprise, and (*b*) closeness of award to personal effort. The basic problem to keep away from is briefly stated as ". . .employees count on the continuation of the (cash) payments and each successive payment seems to have a less stimulating effect" (26).

Is There a Legal Limit?

An outstanding example of "incentive management" is Lincoln Electric. The incentive earnings are not profit sharing according to the company's point of view. In effect, they are double incentive, being paid on earnings from piecework and awards for suggestions. Eighty-five per cent of the employees are said to be on piecework. This combination of incentives has produced phenomenal results. "Prices have been reduced by about 50 per cent and wages have been double those normally paid in industry" (27).

The foregoing quote is from the Congressional Record citing that Lincoln Electric won its case before the United States Tax Court after 10 years of litigation. This outcome clearly points to progress. It contradicts the demands by government for additional taxes and renegotiated prices, and the charge of a Treasury agent that "A man who works with his hands shouldn't be paid as much as $5000 a year" (28).

Cost-Reduction Plans

Clearly, Lincoln Electric's plan provides incentives for cost reduction even though the monies paid come from profits. In contrast, like the usual wage-incentive plans, some plans pay for cost reductions regardless of profits. Two will be described briefly.

The Scanlon Plan was started about 1938 to help pull a company out of the red. As in this case, it is sometimes installed at a union's request. Apparently, it is applied only in unionized plants. The number of installations of Scanlon's plan is not reported. The original application was highly successful and out of it came the book "The Dynamics of Industrial Democracy" (29). The plan sets up a "normal" labor cost and gives "labor the benefit of anything it can save under that 'norm' " (30). The ratio may be revised upward for wage increases or downward for investments made in tooling.

Where possible, the measure used is a "ratio of labor cost to total production value." A basic principle is to pay the incentive to both direct and indirect, and Scanlon prefers to include management. The payments are made monthly as a per cent of the base rate. Experience at La Pointe caused them to set up a reserve of 15 per cent to protect the company against losses when payroll costs might exceed the norm. Another source reports this reserve to be $7^1/_2$ per cent (31).

The other plan is called the "Rucker Share of Production Plan." It has been installed in about 20 plants in the United States and other countries. The plan is based on historical evidence of a relation between payroll and "value added by manufacture." Value added is called conversion by some.

Like the Scanlon Plan, a per cent factor is used but it is computed after material costs have been excluded. The "receipts from conversion" are then set up as two parts of 100 per cent—the company's share and the hourly employee's share. The company's portion applies against all salary, depreciation, taxes, and like administrative costs leaving profit or loss as a difference. The employee's share of value added is matched against the total hourly payroll to determine the incentive bonus as a difference. Here too a reserve is set up to take care of excess costs. The reserve of one fourth is closed out each year. The company takes the loss if it is a deficit.

The Rucker plan does not share the bonus with supervisory and technical people. However, it utilizes the management-employee committee method of administration whether the plant has a union or not.

Both plans emphasize teamwork as the means of gaining security and productivity as the way to increasing earnings. Important gains are shown in the case histories given (31).

Pension Plans

Whether pension plans are wage incentives can be debated. One reason for inclusion here might be that given in the report of the Steel Industry Board. Item (*e*) concludes, "with the knowledge that the economic hazards of life will be at least partially met, workers will be more apt to help sustain consumption spending at a high stable level" (32).

At any rate, their recommendation set off a drive for pensions. On September 29, 1949, Ford contracted for $100 pensions. November 11, 1949, U. S. Steel announced the strike settlement with $100 pensions. November 27, 1949, Secretary Tobin called for a $100 pension at once "for all retired persons" (33). One index of this trend is given by Sumner H. Slichter. "In 1945, there were less than half a million workers under collective bargaining pension plans. By the middle of 1948, there were 1,650,000. At present (February, 1950) over 4,000,000 workers are employed in plants which have these pension plans" (34).

Pros and cons of employee contributions to their pensions were discussed at length. Probable loss of part or all of pensions through inflation or business failure or changing jobs was pointed out. Pressure was exerted to make pensions transferable between certain plants in Toledo. It appears that this pressure will continue either for the individual to remain in one plant as a member of his union in good standing or to have government take over pensions. In this connection, it is reported, "The cost of Old Age Assistance is increasing rapidly—from $430,000,000 in 1939 to $1,300,000,000 in 1949. The tendency of the system seems to be to destroy the incentive to save by promising that those who do not will be supported in their old age by the government" (35).

Executive Incentive Bonus

At the same time, incentive plans have been extended to managements. "Fifty-four per cent of 818 companies in 35 industries responding to an AMA study of executive compensation pay incentive bonuses to their executives. This compares with only 20 per cent of the same companies five years earlier" (36).

There has been a rapid rise too in the number of plans for deferring payments of bonuses. These were set up to encourage ex-

ecutive development and progress within and between companies. A number of these plans are designed to offset the deterrents of progressive taxation in accepting greater responsibilities in management (37).

"The Revenue Act of 1950 has revived interest in stock options as a device for giving executives extra compensation. . .Section 383 frees an executive who buys stock of his company at bargain rates from any tax at the time he exercises the option and provides that any profits derived from subsequent sale shall be taxed only as a capital gain, at a rate of 25 per cent" (38).

Anti-Stop-Watch Rider Dropped

Progress has been made in another direction. On August 26, 1949, Senators Flanders, Taft, and others succeeded in cutting Section 630 off the National Military Establishment Appropriations Act (HR 4146), that was an antiwork-measurement-incentive rider. Similar ones have been attached to Army, Navy, and Post Office appropriations since 1915. Such riders prohibited the payment of any part of the appropriations to any person, firm, or corporation, government employee or outsider, to make time studies, or for any premium or bonus or cash awards except for suggestions that resulted in improvements (39).

This prohibition was overlooked in war production of all kinds. But its elimination has allowed a rapid expansion of work study and methods in many branches of the Armed Forces and some government bureaus.

What's Ahead?

Considering all that has occurred in this period, it is my impression that too many of us look upon incentive plans as cure-alls. Seemingly, each of our detailed approaches is taken as a total solution. Such cannot be true. Assuming employee acceptance, you get high individual performance and skill with an individual incentive plan. You get teamwork with the Lincoln, Rucker, or Scanlon type of plan. You get a sense of "being in business for myself" with profit sharing. You get better methods and creative outlets with successful suggestion plans. To gain all these advantages, it seems to me evident that you have to use all these plans in combination.

Also, it seems obvious that the application of incentive plans can be greatly improved. "In many plants incentives for efficiency could be improved by changes in methods of payment—such as replacing daywork with piecework or other methods of payment by results" (40). This is emphasized by the very low per cent of coverage indicated by the BLS report given earlier.

We should remember also that better incentives for executive leadership are exceedingly important. We need progressive leaders to insure the continued success and development of industry. We must offset in some way the deterrent to incentives of high progressive taxation. In 30 per cent of the companies interviewed (41) there has been some reduction in executive effort. Indirectly, the same thing happens when lack of adequate retirement provisions causes executives to hang onto their jobs. To this Lincoln (42) says, "Since the average span of life is increasing rapidly, funerals do not occur nearly as often as they should for the best progress of most companies."

What to expect in the future is anybody's guess. Currently, we have the highest wages and highest taxes in our history with returns to stockholders so cut down that investments in business are discouraged. Consequently, more and more pressures are placed on ways to increase profits both to pay the bills and to provide for plant replacement and expansion at inflated costs. Therefore it appears likely that the use of incentives will be expanded:

1 To reduce the relative costs of both manufacturing and distribution.

2 To increase the productivity of our available facilities.

3 To stimulate the growth of ideas for methods improvement and for new products.

4 To encourage the progress and development of individuals toward more job satisfaction.

5 To restore teamwork within an organization for the greater security of all employees.

"Particularly in this day and age when the operations of the economy depend so largely upon the policies of business, trade unions, and the government, a responsibility rests upon managers, trade union leaders, and public officials alike to encourage more production and thus make possible a continued advance in wages" (40).

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Industrial-Plant Operation

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Plant Location

THE same fundamental influences that always have affected plant location were operative during the past decade: Proximity to raw materials and markets; availability of fuel, water, labor, transportation; cost of transportation.

Dispersion, in the interest of defense, has been an additional influence. Some plants were located inland during World War II to get away from the danger of attack by carrier-based and suicide planes. More recently, the danger of atomic attack has stimulated plans for dispersion.

Labor difficulties in some areas have influenced both dispersion and decentralization. So has the availability of labor in rural areas, with weight given to the fact that continuing interest in the land on the part of workers is good during periods of prosperity as well as depression.

Availability of oil, gas, and other raw materials has had great influence upon plant location in the South (east and central) by the petroleum and chemical industries. So has the availability of water for both transportation and processing determined plant sites.

Hydroelectric developments have dictated the location of certain plants whose operation demands heavy consumption of electrical energy. Weather conditions have influenced the location of some aircraft factories.

Availability of the right kinds of skills was the controlling factor in locating some government plants during the war, even though the areas already were experiencing full employment. There has been a trend toward outlying and rural areas to find parking space for employees.

The influence of concessions made by communities has lessened. There is a growing understanding of the economics of plant location as contrasted with the bidding of communities. On the other hand, there are zoning restrictions which have some effect upon location, though they are usually local in character.

Local Conditions Influence Location

The characteristics of the community itself may exert controlling influence. In addition to an available quantity of labor at fair rates, what of the turnover rate? Is the community large enough to absorb the employees who must be brought in? Are there enough schools, churches, and housing? Are there vocational schools? How much debt does the community carry? Are there good sites with conveniently located highways and railroad lines? Is there a good airport? Can conditions meet the special requirements of the manufacturer, such as room to test planes, and location for testing television?

Plant relocation has come about in some instances because of the desire of management not to be responsible for the employment of more than a reasonable percentage (about 15 per cent) of the working force of a given community.

Changing from one big unit to a number of small units, each largely autonomous in operation, has permitted increased flexibility in the selection of location and site, especially because smaller working forces are required.

A study by the economics department of the McGraw-Hill Publishing Company, in 1948, showed migration of industry from the New England and Middle Atlantic areas into the Great Lakes, Southwest, Southeast, and Far West areas. The Far West area remained about constant. The movements were quite slow.

In 1951 an analysis by the same organization indicated a continued shuffling of relative positions by areas. This study compared "the per cent of fast write-offs for new plants since Korea" with "the per cent of total manufacturing in 1947." In general, the industrial migration discovered in the 1948 study continued.

Industrial Plant Buildings and Services

The buildings which house plant operations have changed during the past decade. Considerable additional impetus was given to the policy of building the plant around the process—a policy that was not new, but which had not been as widely adopted as it should have been. Chemical plants afford typical examples. There is a growing tendency, when climate permits, to build such plants "in the open," without enclosing walls but usually under roofs. Airplane plants, with their wide spans and other features, furnish another example.

Probably the most marked trend in structural design is the increased development of rigid or semirigid frames in steel as well as in concrete. Economy has resulted from reduced sizes of structural members and simplification of bracing. Welded trusses and other structural elements have been developed.

The desire to produce roof spans as large as possible, with unobstructed vertical clearances, can be interpreted as in the interest of flexible and efficient production. Wider spans, fewer columns, and highly adaptable service installations are characteristic of today's new plants. Spans of 300 ft in steel and 100 ft in con-

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crete have been constructed. Up to 60 ft is common, but the extreme spans are relatively few, because of the high cost. Bays are 40 × 80 ft, 60 × 60 ft, 25 × 100 ft.

Development of the tier-lift truck permits high stacking and, therefore, demands greater vertical clearances; the modern plant is providing them.

There has been some indication of a demand for "surge-capacity units," that is, storage space for important items that come into the plant, and space needed at critical points along the production line. Supplying surge capacity may involve the installation of additional process equipment when the material must pass directly from one process step to another.

Single-Story Plants Predominate. There is almost universal adoption of the single-story plant except where processing equipment requires multistory housing. This includes the use of multistory buildings to take advantage of the gravity flow of materials.

The war brought about the emergency use of reinforced concrete for single-story buildings. By adopting the principle of the arch in combination with the beam, the requirement of reinforcing steel was reduced to less than 3 lb per sq ft of ground area.

Construction costs favor the single-story structure. Using an index cost figure of 100 for a single-story structure, multiple-story plants, for the same floor space and of the same type of construction materials, would cost as follows: 2-story, 105; 3-story, 110; 4-story, 115; 5-story, 120. A well-known engineering construction firm is authority for this statement.

It was necessary during World War II that nonessential features be omitted from industrial buildings. The tendency to omit them continues. Functional characteristics are depended upon to a great extent to give buildings the right appearance. Landscaping is not being overlooked. Architects are using such devices as glass blocks, small brick areas, and distinctive entrances to give plants good looks. Glazed or colored ceramic-tile walls and terrazzo floors are used for the same purpose.

How Much Natural Light? Many plants emphasize the use of a maximum amount of natural daylight. Windowless, or blind, plants, have not gained greatly in popularity, though they have by no means disappeared. All such plants must be air conditioned, and must depend entirely upon artificial light. The fact that all production is carried on under the same conditions of temperature, humidity, cleanliness, and light is advanced by proponents of windowless buildings as a distinct advantage. It also has been responsible for the descriptive term, "controlled-conditions plants." Even the controlled-conditions plants are no longer windowless as a rule. Vision strips, at least, are being supplied. In the chemical industry, however, strictly windowless construction has increased.

Ease in Servicing. Service features have increased in importance in the minds of plant designers. Planning for greatest flexibility of use, one plant (and there may be a good many others) installed service-line grids embracing the entire manufacturing area, so that equipment layout can be changed easily. In this plant, if a hole is bored at any point in the wooden main floor of the manufacturing building, it will come out not more than 10 ft from a set of pipes carrying a total of 12 different services. There are two groups of piping, the first carrying house water, cold city water, steam-condensate return, cold soft water, and hot soft water; the second including low-pressure air, high-pressure air, city gas, hydrogen, nitrogen, and oxygen—an extreme case, maybe, but illustrative.

Definite moves have been made to remove distribution equipment of all kinds from production areas. Sometimes underfloor ducts are provided to contain the distribution systems for air, power, water lines, and other services. These distribution lines sometimes are located among the roof trusses. In either case, it is not difficult to bring leads out very close to the point of use. Equally extensive is the use of bus and trolley ducts for convenience in tapping power and lighting circuits. Unit substations for power distribution have become fairly commonplace. Mezzanines and roofs are used to support compressors, pumps, substations, and other items of service equipment.

Both underground and overhead passages have been adopted to enable employees to get into and out of the plant. Passages lead from gates, highways, or parking places to production areas. They may be equipped with benches, lavatories, toilets, and coatrooms, and are likely to contain the time clocks. Service and sanitary facilities sometimes are located on mezzanines.

Air-conditioning units frequently are located on roofs. Plants are sectionalized, so that the area served by a specific unit is limited.

Corrosive and dangerous process wastes are conveyed to neutralizing plants. Fumes are carried away from plants before being released to the atmosphere and may be neutralized. In the same vein, attention is being given to the elimination or suppression of factory noises that would cause dissatisfaction in the community.

Walls. There is increased use of insulated metal panels for walls. The same is true of asbestos cement and poured-in-place or precast-concrete walls. Use of cinder block in side walls also has increased. The trend is toward more glass and glazed tile, while the use of masonry has declined. Corrugated asbestos sheets for siding have had wider adoption. Aluminum, as well as steel, sashes are popular. Plastic windowpanes are being used more extensively.

Floors. Floors are being built to accommodate loads up to 60,000 lb, the capacity of some of the newer industrial trucks. They are becoming functional; for example, white cement floors are used in some plants to reflect either natural or artificial light to the underside of assemblies, such as airplanes. Asphalt tile is a floor material widely used, particularly in offices.

The following description indicates the extent to which floor construction can become complicated: Here is a floor designed for resilience, vibration absorption, and moisture resistance. The base is 3 in. of tar concrete, which supports a $1^1/_2$-in. layer of tarred sand. Embedded in the sand are 3-in. creosoted square-edge timbers on which is laid a $^3/_4$-in. salt-treated pine subfloor. It is covered with a layer of waterproof paper, over which is laid a $^{25}/_{32}$-in. hard-maple floor.

Roofs. All the established roof constructions continue to be used. Built-up roofs have increased in popularity. One type consists of a welded-steel deck, covered with $^1/_2$-in. fiber-board insulation which, in turn, is covered by alternate layers of tar or asphalt and waterproof paper. The use of gypsum plank in roofs has increased. Plastic skylights and roof coverings are coming into use.

Heating. Unit heaters continue to be the predominant type of heating equipment. The heat-blast system is used to some extent. Blowers change the air, and the frequency of change can be arranged to take care of employee comfort as well as safety. Radiant heat is on the increase, particularly in the small plant. The high-pressure hot-water system is gaining popularity.

Lighting. Fluorescent lighting, with 50-ft bulbs not uncommon, has been the most popular type of lighting. Sometimes it is used in combination with high-intensity mercury-vapor lighting. Becoming popular is the fluorescent-mercury-vapor bulb, available in 400 and 1000-watt sizes. A new general-purpose mercury-vapor lamp, with built-in reflector, is designed to require no cleaning during its life. Incandescent bulbs also are popular and cold-cathode lighting has made considerable progress. Photocells to turn on lights have had only limited adoption.

Worker Comfort and Health. That the plant should be a good place in which to work now is a philosophy universally accepted.

There is considerable conviction to the effect that production costs are directly related to worker morale.

Germicidal lamps have become a feature of worker protection. They are used in cafeterias, first-aid rooms, lavatories, plant hospitals, toilets, and sometimes in offices.

Moving stairways have made their appearance in industrial plants. They are used largely for the convenience of employees, getting them quickly to and from their work places.

For employee summer comfort, in some plants, water is pumped through the heating pipes over which air is blown. The waste cooling water is pumped over the roof, furthering the cooling process.

Air conditioning has had a remarkable increase in application. Where the product or process requires it, use is a matter of course. But there are many examples of applications to offices for the greater comfort and higher efficiency of white-collar workers. Electrostatic precipitation of dust continues to be used rather widely.

Acoustical ceilings achieved great popularity, particularly in offices, and are used to some extent in shop areas.

In offices and finished areas, suspended ceilings of glass fiberboard, glass foam, and vegetable fiberboard are finding favor. Color for walls, ceilings, and equipment has become accepted practice for many plants. Color codes on piping are common, fostering worker safety as well as more effective maintenance.

Effect on the Small Plant. There has been a natural tendency for leadership in plant design and construction to take place in the larger plants. But the small plants are showing their fair share of originality. A survey of 14 small and medium-sized plants in the general industrial area centered by Chicago shows that steel-frame construction was used in 13, radiant heat in 6, steel sash in all, air-conditioned office sections in 9, color in 13, and parking space was provided by 12. These figures are interesting particularly in view of industry's trend toward decentralization with the consequent use of smaller plants.

Plant Layout

Developments in plant layout have occurred principally in the tools used—boards, charts, models, templets; and in kinds of equipment—workplaces, racks, benches, bins, hoppers, trays, and handling devices.

The flow-process chart is being used extensively. Wherever possible, from the cost point of view, functional has been changed to straight-line layout, because of the improved opportunity to make effective use of handling equipment.

Facilitating layout, and affecting costs favorably, has been the use of standard benches, shelving, and other facilities. Standardized workplaces were developed by some firms. So were other items which are mentioned later in this paper. A further indication of the influence of standards was indicated by the publication, in 1948, of "Proposed ASME Standards for Plant Layout Models."

Color became a tool of layout. On some layout charts or diagrams, where more than one product was carried through an area, products and equipment were matched as to colors. Use of color has been made to designate aisles, storage areas, conveyers, sex of operator. Color has been used, also, where pipe lines are indicated on the layout chart. Layout boards are made from plywood, insulating board, wall board, and glass.

Models. Models of machines, equipment, furniture, and operators are $^1/_8$, $^1/_4$, or $^1/_2$ in. to the foot. They are made to scale, with provision to indicate maximum working floor areas (for table travel, swing, or other movement). Material usually used is wood, though some models are cast of white metal and others are molded plastic. Models of buildings are made of wood or transparent plastic.

Revolving Table. The revolving table for assembly work has found a real place in layout work where the operation permits.

"Modular" Office Furniture. In office layout, "modular" furniture has been introduced. L-shaped desks, with partitions built on, and with integral bookshelves and files, greatly reduce working area in comparison with usual arrangements. The partitions give a measure of privacy not possible with a "bull-pen" layout.

Specific Layouts

A review of actual layouts will serve to indicate developments in methods, tools, and equipment. It will show how these items have been combined in specific cases to advance the art of plant layout.

Case 1. The layout board is sectional. The entire floor area, to scale, is blueprinted and mounted on plywood. Aisles, doors, and floor construction are shown. Another copy of the same print is cut into sections and mounted on plywood. The cutouts, or templets, are stapled to the sections, as positions are determined. Individual sections can be lifted off the master print and taken to desk or into the plant for study and revision. When all are in position on the master print, the layout can be studied in its entirety.

Case 2. Three-dimensional layout with scale models is used on the basis that "if it fits in the layout it will fit in the shop." Models include machines, conveyers, benches, columns, toteboxes, floor trucks, and operators. The layouts are made on work sheets ruled in $^1/_2$-in. squares and laid under glass. When finished, the layout is photographed in a series of matched photographs, care being taken to avoid perspective. The composite of photographs is enlarged. By the procedure described, there are no drawings, and there is no measuring. Moving the models is all that has to be done.

Case 3. The models representing equipment are blocks of wood, labeled, and sized to show operating areas required by equipment. A color code designates working areas, stock, aisles, machines, conveyers, supervisory offices, concrete, grounds, outside walls. Red thumb tacks indicate men, yellow tacks women. Completed layout is photographed.

Case 4. Models are made of wood, $^1/_4$ in. to the foot. They give approximate shapes of the equipment. A color code is used. Aisles are shown by white paint. The base is insulating board.

Case 5. Standardized workplaces characterize this layout. Equipment is of the all-purpose type, so that usefulness is not limited to one product. Advantages are conservation of floor space; reduction of direct-labor cost; forced adherence to principles of motion economy and work standardization; reduction in time required to set up the production line or department. There are places for necessary items only; there is no space where junk can accumulate.

Workplace design is based on this principle of motion economy: Use the smallest body member possible to move the shortest distance. To the standardized benches are attached electrical lines and air lines, arranged to be coupled to electrical and air lines of adjacent benches. Benches are supplied with supports or stands for material containers, the material containers, tool supports for prepositioning tools, and clamping devices to permit easy mounting or disassembly of the items just listed.

The following principles are used in making the layouts: Use lip trays, bins, and hoppers for positioning material. Preposition tools for quick and easy use.

Case 6. Scale models of a three-story building and of machines, benches, and other equipment are used. The building model is made of pine, plywood, and pressed wood. It contains both floors and partitions. Layouts can be made a floor at a time. To get a top view of any floor, lift off everything above.

Case 7. The floor area is laid out on cross-section paper which is stapled to panels of wall board. The paper is black with white graph lines; the distance between the lines representing 1 ft. Columns, aisles, and permanent installations are marked by strips of red, stapled in.

To this area layout are added templets of machinery and other equipment. The templets are of white cardboard, marked for equipment identification with black ink, and they are stapled in. The completed board is photostated, and the negative print (usually reduced $^1/_2$) is used so that equipment and graph lines will show in black, and the lettering and background in white.

Case 8. Magnet-cored plastic templets are used. The cross-hatched floor plan is backed by a ferrous plate. Information descriptive of equipment is contained on the templets. When the layout is ready for printing, it is placed over a dry-process sheet of sensitized paper. The exposed sheet is developed over an ammonia vaporizer.

Case 9. Transparent plastic buildings are used. Etched into the floors are 1-in. squares, each representing a 4-ft × 4-ft area. Models are placed. Photos of floors form permanent records of layouts.

Case 10. Scale models are placed on glass sheets and are held by an adhesive that will not harden. With models in position, the layout is turned upside down. A transparent acetate grid sheet is placed on the bottom of the glass sheet. Templets, matching the models, are stuck to the acetate grid. The acetate sheet is then placed on sensitized blueprint paper for copying.

Standardization. Special mention should be made of facilities standardized by one company in the interest of good layout. Here are the items: Totebox dolly, service dolly, instrument storage dolly, pipe and chain railing, office partition, expanded metal partition, factory writing stand, rack for wire wheels, underbench rack, cloakroom rack, stock rack, tool-crib rack, warehouse rack, bench, rest-area bench, laboratory work bench, chair, utility cabinet, janitor's truck, and box locker.

Traveling Stockrooms. There has been a revival and extension of the use of traveling stockrooms, an arrangement that strongly influences plant layout. One user has said that when the nature of the manufacture permits, traveling stockrooms can be incorporated in the layout to eliminate manual transportations; minimize handling at the workplace; save floor space; assure steady flow of work, guarantee flexibility of operation; facilitate layout changes; avoid congestion; aid good housekeeping; and give stock-handling economy. Stock, finished-unit, storage, and shipping conveyers can be used.

Service Lines. Both underfloor ducts and overhead carriers for service lines have been given much attention in connection with plant layout. In one case, service equipment has been suspended from the ceiling below the production floor. Pipe lines, ventilating-duct work, and power duct are included. Blowers are located on a foundation at the basement floor level.

Materials Handling

Broadly descriptive of progress in materials handling during the past 10 years are the following operations: Palletizing, stacking, sideloading, lifting, automatic integration, automation, self-aligning, telescoping, and positioning. These will be referred to in the following paragraphs.

Conveyers. The use of conveyers of all types has grown tremendously. Outstanding conveyer developments and applications are cited to indicate directions taken during the 10-year period.

Of particular importance is the practice of using conveyers in handling between-production operations. Short-flight gravity conveyers as a rule are employed.

"Automation," a coined term, describes the joint process of machining and handling parts. The automotive industry is most active in this development. A series of operations is performed on a part, such as a cylinder block, by machines or machine units placed as close together as possible and connected by short runs of conveyers. The combination is called a "transfer" machine. Everything is automatic—clamping, machining, unclamping, changing of positions, and moving the work.

The guided pallet conveyer combines power and gravity flow. Basic sections in short lengths are bolted together, permitting pallet travel over long or short distances. A guide track runs along the center line of each section. A caster, mounted on a vertical shaft attached to the bottom of each pallet, fits into the guide and keeps the pallet moving evenly on the roller tracks. The power, or pusher, sections are provided with motors which drive chains equipped with cleats that engage the pallets and push them along. The power units, located at key points in the line, establish a time cycle. They raise the pallets to gain elevation, and thereby index the pallets in the gravity line.

Power-and-free conveyers, properly integrated, provide automatic travel between any two points. Free loops permit both type and rate of production to vary from zero to full capacity at each station, without affecting any other station, and without need for adjustment. Carriers are accepted or rejected automatically. Control of movement over the system is by means of limit switches, two-way power-operated tongue switches, and manually operated stub switches. Variable-speed drives are powered by electric motors.

For some kinds of bulk materials, the "zipper" conveyer is appropriate. It is a U-shaped belt pipe line, fitted lengthwise on both edges with teeth that can interlock. It is loaded from the top, continuously "zipped," turned 90 deg in transit, then continuously "unzipped" to unload from the bottom.

Overhead conveyers to carry work through production processes are used much more extensively, especially in plants manufacturing in job lots, such as foundries, furniture, and sheet-steel fabrication plants.

Automatic integration of conveyer systems is increasing. It is accomplished by the application of both mechanical and electronic devices.

Extremely long belt conveyers, now installed in several coal mines, represent the solution of difficult problems of engineering. Lessons learned from these installations may be applied in manufacture.

Some of the individual items will give a good idea of new things developed:

A "turnover" belt-conveyer system turns the belt after it delivers the load, runs the clean surface on the idlers.

Steel conveyer belts, both link-type and sheet, handle hot materials.

Telescoping-belt conveyers provide a flexible, quickly changed setup for variable operating conditions, particularly for shipping.

An automatic pallet loader eliminates hand-loading of cartons. Set up at the delivery end of the packaging line, it loads pallets and then pushes them on a roller conveyer.

A small spiral conveyer, carrying small parts on hangers, is so designed that it can be installed along a path with many turns.

A roller that keeps parts on the conveyer and takes them around curves without side guards.

Pallets and Fork Trucks. Pallets and pallet-handling equipment really came into their own. The use of palletized unit loads expanded tremendously, resulting in lower labor cost, faster handling and better utilization of space.

Expendable pallets, made of paper, wood, or fiberboard, reached full utility development. So did collapsible basket pallets.

Indicative of developments made in lift trucks and other devices concerned with palletizing are the following:

Stacking crane cuts aisle-space requirements.

Sideloader attachment for lift trucks cuts aisle-width requirement for tiering.

Forkless device for handling pallets reduces aisle space.

Fork-truck device permits use of paperboard sheet as pallet.

Adjustable pallet racks store miscellaneous, uneven pallet loads without excessive waste of space.

There are many other developments in trucks, the tendency being to adapt the fork to the specific job, although in some instances the devices do not use forks. In other cases, clamping attachments are arranged to dispense with the pallet. In addition, there have been improvements in truck performance, including higher stacking, faster hoisting, and better maneuverability.

A forkless device handles cartons containing bulky and heavy product by inserting a hook under the top of the carton. Various clamping attachments for fork-lift trucks handle cartons, barrels, drums, bales, and rolls without pallets.

Some containers, such as nesting tote boxes, are self-stacking.

Steel skid boxes, some of which open at the bottom or side, convert themselves to self-feeding hoppers.

Other Noteworthy Developments. Automatic stacking of boxes is being practiced. The loaded boxes are lifted from a conveyer by the stacker and put into the appropriate stack, as determined by an inspector who operates a push-button control. When the stack is filled, it is automatically moved to a floor track for removal by truck.

Positioning equipment minimizes manual handling at machine or workplace. Examples are hydraulic elevating tables, power-operated upenders and welding jigs. Advocates claim savings in floor space, equipment, and manpower.

An overhead crane designed to follow an S-curve runway is in successful operation. A new design of pneumatic-tube system has proved effective. It is fully automatic, permitting transmission from one station to any other by dialing the proper number.

Yard handling makes greater use of trucks and truck-mounted cranes. The straddle truck, originally designed for handling lumber, has been adapted to handle many more materials in bundles and on pallets.

Bridge plates, to span the gap between loading dock and carrier conveyance, have been improved. Some are equipped with brackets for truck handling and are made of aluminum or magnesium to minimize weight.

Maintenance

The most significant advance in maintenance is the recognition given the function by management. This has been illustrated by the enthusiastic reception accorded a plant-maintenance conference and exposition launched in 1950, and continued in 1951 and 1952. Audiences as large as 2000 have attended the sessions, and as many as 14,000 people have visited the 4-day exposition of maintenance equipment, tools, and supplies held concurrently.

To these gatherings have come not only the top men in maintenance, but a good sprinkling of production management as well. Apparently there has been no hesitancy on the part of management to endorse the trips and pay the expenses. Moreover, attendance has been national in character.

As a result of these conferences, important additions have been made to maintenance literature. They consist of the conference transactions, which contain the papers and addresses plus the questions and answers presented at the 3 days of sessions of the three conferences.

The conferences and their transactions have stimulated many publications serving industry to increase their maintenance content. Thus still more additions to maintenance literature have been made.

Indicative, also, of recognition, is the participation by both maintenance and production executives in the 3-day maintenance seminars carried on for the past three years by the American Management Association.

A third indicator is the increased interest in maintenance on the part of consulting engineers. Some of them report extensive and increasing use of their services by industry.

High prices for materials and equipment, elevated labor rates, increased taxes, and scarcity of manpower have built up a consciousness of the importance of maintenance costs. By their attempts to determine costs and to compare their costs with the costs of other companies, plant engineers, maintenance engineers, and other representatives of management have discovered that there is no agreement even within an industry as to classification of costs in line with definitions of specific maintenance responsibilities. There has grown a determination to remedy this shortcoming.

Equally recognized for its importance is preventive maintenance, with its obligations to set up and follow definite inspection routines; and to establish, keep up, and analyze maintenance and repair records.

Planning and scheduling of maintenance work is recognized as another essential operation in the management of maintenance. A great deal of study is being put in on repair order forms and other paper work.

The place of maintenance in the organization also is receiving the attention of top maintenance and administrative executives. The standing of the function and, therefore, the people responsible for it is getting higher. The function is approaching professional status. In the author's surveys of what maintenance executives are interested in, maintenance organization and management has been found close to the top of the list. These surveys show that they also are interested in the following: Inspection procedures and frequencies; Planning and scheduling maintenance operations; Maintenance costs and their control; Training people for maintenance work; Incentive pay for maintenance workers; Records and reports.

The reader should refer to other parts of this report on progress in plant operation for additional information on maintenance—for example, service features in building construction, and plant layout. Not mentioned elsewhere are "area" rather than "central" maintenance (of interest to large plants); replacement of light bulbs by groups on the basis of hours burned rather than individually upon final failure; developments in safe and mobile scaffolding for overhead work; and new methods for preventing corrosion.

Much greater recognition has been given to inculcating in maintenance executives a recognition of the importance of good personnel relations.

Mentioned elsewhere, but deserving of reiteration here, is management's recognition of the importance of good plant housekeeping because of its effect upon worker attitudes. The maintenance forces are carrying the responsibility for housekeeping, and are paying much attention to rest and locker rooms, feeding quarters, sanitary facilities, painting, lighting, and landscaping.

There still is no clear-cut line between maintenance and plant engineering. To some production management and some maintenance management personnel, it seems important to determine where the line should be drawn. To others, it does not. It is quite likely that there will be much discussion of this subject during the next several years.

Project preparation and cost control is another phase of maintenance that is beginning to receive consideration. It is defined by one of its sponsors as "Methods used to prepare request, design, estimate, carry on field work, control costs, and so on, for the general run-of-mill type of new work and alterations that are

encountered in industrial plants of all sizes with the majority of the projects under the $50,000 class." As time goes on this definition probably will be revised in many plants to delete the financial limitation.

There is a growing recognition of the gains to be made from the "industrial engineering approach" to maintenance. Design is the first step to insure buildings and services that will favorably affect efficiency and cost. As we get closer and closer to the push-button plant, the ratio of maintenance forces to production forces goes up. At the same time, the ratio of trained and skilled to unskilled maintenance men goes up, largely because of the complications of electronic controls and instrumentation.

These facts are indicative of the changing character of the maintenance function. They point to a much higher standing of maintenance executives during the coming decade as compared with their standing today, which is above that of 10 years ago.

Purchasing

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Introduction

Over the past 10 years industrial purchasing has been carried on under a series of extraordinary conditions. The first half of this period was dominated by the war-production economy, with its attendant material shortages, priorities, and allocations, and a variety of governmental control regulations. This was followed by the period of reconversion to civilian-goods production, strongly inflationary cost and price trends, and the return to a half-war economy as a result of the Korean action and the defense-production program. In considering the development of purchasing as a management function during these years, the effects of these conditions cannot be overlooked, but there is little to be gained by a rehearsal of the means by which these particular problems were met and overcome. The important thing is to note the more permanent changes or developments which have been prompted or accelerated by this experience.

Most significant is the fact that the trend of purchasing policy in manufacturing industry has been strongly in the direction of procuring a greater proportion of component parts and assemblies in fabricated form from specialized supplier industries, rather than purchasing raw materials for conversion in more completely integrated operations. This is shown clearly in analyses of the purchase list of representative companies in almost every manufacturing field.

The trend toward procurement by purchase rather than by manufacture reflects in part the wartime emphasis on subcontracting. Thus practice was promoted by government contracting officers and war contract regulations for the triple purpose of utilizing existing production capacity to the utmost, saving plant-conversion time, and sustaining smaller and less essential industries which otherwise might have fallen victims to the war-production program. Similar official pressures have been brought to bear upon purchasing policies and practices in the postwar period, largely from social-economic motives in behalf of smaller business and distressed areas, after the urgency of war-production needs ceased to be the dominating factor.

Management Trends Affecting Procurement

But while government policy has been the initiating force in this new approach to procurement, more permanent reasons for regarding it as a continuing policy and trend are found with industry management itself. One of these is the greater attention now being given to "make or buy?" analyses and decisions before undertaking the manufacture of a new part or component, or providing capital equipment and facilities for such purposes. Today, a careful survey is made of all factors. This frequently leads to the "farming out" or purchase of parts that formerly were manufactured, perhaps at an economic disadvantage, within the plant. In principle, this always has been a factor to be considered in procurement, but decisions frequently have been made without full analysis, or have been influenced by the urge for expansion. Mandatory subcontracting indicated a more selective and objective approach, and objective analysis has established this policy on its own merits.

Closely related to this is the growing tendency for management to concentrate on the major type of operation to which the company is primarily devoted, thus putting into effect the same principle of specialization which is being sought and found successful among its suppliers, rather than seeking self-sufficiency in vertical integration.

A third management trend contributing to this type procurement is the policy of growth by diversification of end-product, often in the form of an organization by product divisions, as a prudent economic hedge. This sort of outlet for expansion again indicates the wisdom of placing greater reliance upon suppliers and on purchasing to simplify the final fabricating operations.

There have been some very interesting corollaries to this policy. One of these is the pattern of industrial development in the relatively new and rapidly growing industrial areas of the Southwest and Far West. As the major industries (e.g., steelmaking, automotive, aircraft, chemical, and so on) entered these areas, they became the nucleus for active development of a wide variety of specialized supporting industries, thus bringing the sources of fabricated components closer to the new points of need.

No accidental development, this is the result of definite and consistent purchasing policies. An example is the West Coast purchasing program of the Ford Motor Company, instituted in 1947, in connection with parts procurement for the Ford plants at Richmond, Long Beach, and Los Angeles, Calif. At the time the vast majority of parts had to be purchased from eastern manufacturers and shipped by rail across the continent. This involved transportation costs and amounted to from 5 to 30 per cent of product value, plus the necessity of maintaining a considerable inventory in transit. There was no practical alternative, however, since nearby suppliers had neither the capacity to furnish the desired items in the required volume, nor competitive pricing, even without the added costs of transportation from established eastern sources.

Developing Source of Supply

In February, 1947, Ford's director of purchases announced the objective of developing a $50,000,000 annual purchasing program among West Coast suppliers to eliminate the long and costly

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haul. Only a small fraction of the volume was available then from sources in the area The means used to attain the objective were:

1 Establishment of a permanent office at Los Angeles, where a special representative of the director of purchases contacted approximately 3000 potential suppliers during the first year of the program, not as a buyer but as a liaison and development agent.

2 A permanent display of purchased parts which prospective suppliers could study in relation to their own facilities.

3 Technical assistance.

4 Underwriting of necessary plant expansion by the assurance of long-term contracts.

5 Temporary price subsidies, in some cases up to the amount of the transportation differential.

After two years of intensive work, this purchasing program was halfway toward its goal. Ford was able to purchase $27,000,000 worth of its requirements for the West Coast plants from nearby sources, and the process of industrial development was well under way. This was distinctly a purchasing project, initiated and executed by the purchasing department. It can be duplicated in the experiences of other leading companies.

Within the field of company organization, this trend toward procurement of materials in more highly fabricated form has enhanced the importance and the position of the purchasing function because the proportion of the sales-income dollar expended in purchasing has increased substantially. Basically, it is probable that the ratio of cost of material to cost of manufacture in the finished product is relatively stable. However, for the individual company, when the supplier's manufacturing cost is added to the cost of material in the fabricated part being purchased (at the same time that these manufacturing expenses are delegated to the supplier from previously existing ratios), the greater share appears as a purchasing expenditure and a purchasing responsibility. For example, 10 years ago in the automotive industry, purchases represented about 52 to 54 per cent of the sales-income dollar, approximately the general average for all manufacturing industry; today, representative companies in that field report purchases ranging from 60 to 62 per cent of income. Similar trends are reported in other fields. If and as the manufacturing industry becomes more and more an assembly operation based on specialized production, this ratio will continue to increase.

Materials Control

The effect of this is to focus management attention more sharply upon the function of procurement as the purchasing responsibility assumes greater importance both policywise and costwise. It has accelerated greatly the recognition of purchasing as a management function knowing a large and direct bearing on company profits, beyond its original and continuing function as a service supporting the manufacturing operations.

This, together with wartime experience in scheduling and controls, is broadening the entire concept of materials management in the industrial process. Greater attention is being given to materials control, sometimes as a separate and specific responsibility in the organization, and sometimes within the framework of the existing purchasing department. It would be academic to argue the relative merits of either method, or the logical line of development to be followed. As in most management problems of this nature, the executive or department with the capacity and initiative to see and accept the larger responsibility is likely to resolve the issue and dominate the situation. In many cases the purchasing department, which already is charged with the procurement of materials and frequently the supervision of stores, has acquired a more comprehensive responsibility of materials control. The important thing to note is that there is emerging a distinct, inclusive function of materials management, of which the actual purchasing is only one of many facets, including inventory policy and control, materials control, traffic, and stores. This whole combined responsibility must operate at the top policy level.

Published statements by the managing executives of many leading companies provide testimony to the wide recognition and acceptance of this changing emphasis. There is further emphasis in the increasing number of purchasing officers who are designated by the title of "manager of materials" instead of the conventional "purchasing agent" or "director of purchases," which has ceased to be completely descriptive.

The job of the person in charge of this function, whatever his title, likewise has changed, since there is a definite requirement of administrative skill in the management phase, in addition to the buying skills which always have been associated with purchasing. Even in the smaller companies and purchasing departments, where the two responsibilities rest in a single individual, it is recognized that the greatest services to the company as a whole are achieved, and the fullest potential of the operational phases of buying are realized, when the administrative and management aspects of the job are given opportunity and attention.

Materials management is still in the exploratory and development stage, and the greatest progress in the line lies ahead. Much progress has been made in the organization of knowledge and experience, training and education, for purchasing handled within individual departments, through the professional associations in the field, and in management education. This phase of the procurement function is approaching the status of an engineering science, with well-established principles and techniques. Many companies have compiled purchasing-department manuals, covering policies, procedures, or both, which regularly are used as a guide to standard practice. More than 100 courses in purchasing are offered regularly today in schools and colleges of business administration, both at the undergraduate and graduate levels. It is significant that these courses generally stress the administrative phase of the function, and are accepted as a standard part of the management curriculum rather than from a vocational training viewpoint.

Purchasing Organization

Centralized purchasing, in the sense of setting up procurement as a separate department and a specialized functional responsibility within the organization, continues to gain in acceptance and practice. This is true not only in industry, but in the management of local and national governmental activities, public and private institutions, hospitals, schools, and a wide range of other enterprises in which goods must be acquired for any sort of operation. Any period of material shortages, such as has prevailed during the past decade, gives impetus to this development. Other factors already noted have added to this progress and are given in permanency.

With the new emphasis of purchasing administration, the actual buying may be regarded as an operational activity under the jurisdiction of the purchasing manager, but tending to become decentralized in proportion as over-all management is decentralized. This is evident particularly in widely spread operations and in the multiplant type of organization, characteristic of even medium-sized industries today. Here, the tendency is away from the strongly centralized buying unit at headquarters. The function of the central purchasing office currently is regarded as having the responsibility for policies, administration, methods, and control. The actual buying (except for major contracts covering requirements common to several plants or

divisions) is done through divisional purchasing offices having a considerable degree of autonomy in judgment and action and is closely tied in with the respective division managements that they serve. This is true regardless of the geographical location factors that may be involved.

An example of this is the purchasing organization of the Major Appliance Division of the General Electric Company, which is currently bringing together the manufacturing operations previously carried on at eight scattered locations, with management headquarters at Bridgeport, Conn., into one large, consolidated location known as "Appliance Park" at Louisville, Ky. Physically, this is a process of centralization. In the purchasing organization at this project, there is a manager of materials and purchasing (a newly created position) for the over-all Major Appliance Division. But each of the five product departments—Range and Water Heater, Electric Sink and Cabinet, Home Laundry Equipment, Household Refrigerator, and Room Cooler Departments—has its own purchasing agent and staff.

In announcing the buying policy and procedure to vendors at the opening of Appliance Park in March, 1952, the manager of materials and purchasing stated:

"In general, each of our departments will purchase materials and components to take care of its own production requirements. Where similar materials or possibilities of standardization make it advantageous to consolidate purchases, they will be co-ordinated by the Materials and Purchasing Department of the Major Appliance Division."

This example illustrates the principles of centralization and decentralization that are going on concurrently in the purchasing field as well as the emergence of the materials management function, and the emphasis on administrative phases of purchasing in modern practice. It calls attention to the factors of consolidated (volume) purchasing, standardization, and unified responsibility, which are the basis of centralized purchasing. It also more closely identifies purchasing with the particular manufacturing operation which it serves, which is the basis for decentralized buying.

This type of decentralization or divisionalization, identifying the buying operative closely with individual plant management, but without sacrificing the advantages of centrally administered policy and control, is eminently logical, since it places the purchasing action physically at the point where purchased materials are needed and will be used. Plant and division management are responsible for the efficient, economical, and profitable conduct of the plant. Purchasing is responsible for approximately 50 per cent or more of product cost. The two responsibilities cannot be separated.

Cost Factors

The purchasing department always has been expected to be cost-conscious, to explore markets, to schedule quantities, and to adjust its decisions according to economic fluctuations and trends so as to satisfy the company's requirements at the most economical and advantageous terms. The increasing purchase costs and the larger proportion of fabricated products purchased have added both importance and opportunity to this cost-saving aspect of purchasing responsibility. The greatest recent advance in purchasing technique, which is definitely a development of the past 10 years, is the evolution of planned programs and methods for reducing costs and obtaining better values in procurement. Some of the methods employed are not new, nor are they peculiar to purchasing, but the organized application of these principles and the scientific approach constitute tremendous advances in purchasing practice.

The underlying thought in this technique (which is variously known as purchase analysis, value analysis, or simply cost reduction) was arrived at independently in a number of different companies, and has gained widespread acceptance and application through generous interchange of ideas and experiences in purchasing circles. Basically, it is a new examination of the cost factors in purchased products and what can be done about them.

Purchase cost of a required item is conceived as being made up of two layers. First is the intrinsic cost to the manufacturer or supplier, including materials and the cost of manufacture. This properly has been regarded as a fixed, inviolate, rigid price base which could not and should not be penetrated. The buyer's only recourse at this stage was to find a manufacturer having lower costs. Upon this base are superimposed a variety of more flexible factors—the possible economies of volume production, distribution costs, pricing policies, profit margins, and temper of the market. These are subject to negotiation, and traditionally this has been considered the area in which the purchaser could maneuver legitimately to secure lower costs and higher values.

The new technique of value analysis takes a second look at the basic layer to see whether costs are what they are because of the requirements stated in the order. It also seeks to find whether purchase costs can be lowered by reducing the production cost—through changes in design, substitution of materials, or alternative methods of production—while still procuring a suitable product to serve the intended purpose. This procedure does not question the supplier's cost, nor attempt to shave his margins. In effect, instead of accepting a requisition and seeking a supplier, the purchasing agent first scans the requisition and asks himself, "Are we asking the supplier to furnish an unnecessarily costly product?" If his study reveals that basic costs can be lowered, the savings thus achieved are not at the expense of the supplier; they are fundamental and repetitive, and usually far outweigh the savings that might be obtained by market exploration and negotiation (which are still open to him).

A typical check list used in such an analysis includes the following questions:

Does use of this part contribute value to our product? (Example: A small condenser, used across relay contacts for arc suppression, was found to be unnecessary with alnico magnets to provide snap action. The condenser was eliminated. Unit cost reduction: 10 cents to 0, a 100 per cent saving.)

Is cost of this part proportionate to its usefulness? (Example: A spacer hub seemed unduly expensive, as a result of undercutting needed to reduce weight. Substitution of an aluminum hub, without undercutting, provided identical performance with even less weight. Unit cost reduction: 90 cents to 20 cents, a 78 per cent savings.)

Does this part need all its features? (Example: A stainless-steel disk used in a dispensing machine was specified chamfered on one side. The chamfer contributed nothing to value or utility, and was eliminated. Unit cost reduction: 18 cents to 5 cents, a 72 per cent saving.)

Is there anything better for the intended use? (Example: Mica stock used for insulation was changed to Micalex. The molded contour resulted in a more rigid mounting. Unit cost reduction: 4 cents to 3.4 cents, a 15 per cent saving.)

Can a usable part be made by a lower-cost method? (Example: A hub assembly formerly designed and made as a two-part riveted or staked assembly, was redesigned as a one-piece casting. Unit cost reduction: 30 cents to 10 cents, a 67 per cent saving.)

Can a standard part be found which will be usable? (Example: A stud contact formerly made to special design was replaced by a standard stock design available at about one half the cost. Unit cost reduction: 2.7 cents to 1.4 cents, a 48 per cent saving.)

Is the part made on proper tooling, considering quantities used? (Example: A stainless-steel weld nipple, used in relatively small

quantities, was made by machining away part of a standard fitting. Volume requirements increased to the point where this method was reviewed, and changed to production on an automatic screw machine. Unit cost reduction: 20 cents to 5 cents, a 75 per cent saving.)

Does quoted cost reflect reasonable material, labor, overhead, and profit? (Example: Price of a steel dowel pin, made to special design and specification, seemed excessive. Consultation with the manufacturer on details of the specification, manufacturing process, and inspection resulted in eliminating some wastes of material and labor. Unit cost reduction: 0.3 cent to 0.2 cent, a 33 per cent saving.)

It will be noted from the examples cited in the foregoing that this type of cost reduction must be generally effectuated through design and manufacturing departments. It is not the province of purchasing to authorize changes in materials, design, or production methods upon its own responsibility. But it is the province of purchasing to suggest and initiate such changes. In almost every instance where such an intensive program has been carried on, it has centered in the purchasing department. This department occupies a unique and strategic position to facilitate these changes because of the inherent cost consciousness and cost responsibility of purchasing, because only this department has the opportunity to scrutinize every item of the company's requirements; also because purchasing is aware of the importance of repetitive savings that might seem wholly insignificant to other departments in relation to total product cost.

The examples cited are small, individually. Yet, when multiplied by the thousands of such parts that are required in a year's operation, they represent an annual saving of a quarter-million dollars, year after year, to the company from which these instances are taken, and this is but a minute part of the total program. The percentage figures are significant. They are fantastic, yet commonplace. Applied to the potential of the total purchasing program, they can become very impressive. Another company (Westinghouse), which conducts a similar program, reports that savings of this nature amount to more than 4 times the total cost of operating the purchasing department—a handsome dividend made possible by the application of this new purchasing technique. At the Ford Motor Company, where such a program was inaugurated in 1946, the importance of repetitive small savings is emphasized by a slogan, based on annual production of one million vehicles: "A difference of 1 cent per car in unit cost is a $10,000 error or opportunity."

Public Relations

A review of purchasing progress over the past 10 years would be incomplete without noting the growing appreciation of, and attention to, the public-relations opportunities and values that exist in the purchasing office. This, too, was highlighted during the years when materials were difficult to obtain, when good vendor relationships and co-operation were perhaps a purchasing department's most valuable assets, and when those who had neglected or abused the opportunities to develop these assets found their troubles multiplied manyfold.

Purchasing and sales are the two departments of a company having the most numerous and constant contacts with those outside the organization. Good public relations always have been a part of the salesman's stock in trade; unfortunately, the same has not been true in purchasing. A continual stream of vendors' representatives passes through the purchasing office; to many of them, this is their only contact with the company, and their impressions of the company are based wholly upon the treatment they receive there, and in their business dealings with the buying staff. Beyond the buyer's personal interest in establishing good relationships with his vendors, he has a company responsibility to maintain a high standard in this respect.

This has become a subject of serious and intelligent concern among purchasing men, both individually and through their professional associations. While this is an exceedingly difficult matter to measure and evaluate, the new awareness and higher standards are impressive. Much progress has been made.

In a larger sense, the public-relations responsibility of being a good citizen and co-operative member of the community in which a business operates also must be considered. Purchasing policies can contribute substantially to the good will that a company enjoys. The decentralization of purchasing activities to local levels, and the active development of supplier industries in the areas already cited, are examples of how purchasing can help and is helping.

All of these aspects of public relations emphasize the growing role of purchasing as an integral part of company management and policies, and go beyond the actual buying activities for which the purchasing department exists.

Marketing and Distribution

By NOBLE HALL,[1] PHILADELPHIA, PA.

Introduction

THE decade 1942 to 1952 can be described best as a period of rapid and spasmodic changes. The marketing and distribution picture in the United States has been forced to adjust itself to extremes of situations. It is the purpose of this paper to examine the forces behind these changes and to analyze their significance.

The War Years—1942-1945

The war years, 1942–1945, witnessed the change from partial mobilization to all-out war. As early as 1938 economists, the military, and the diplomatic corps were viewing with increased alarm the gathering of war clouds on our east and west horizons. The development of the totalitarian created new problems in foreign markets, particularly the German industrial influence in Central and South America. The whole struggle for industrial and territorial expansion led eventually to the outbreak of war in 1939 on the European continent.

The natural alignment of interests brought America into the conflict, if only indirectly. The institution of lend-lease, the Murmansk run, the arms and defense activity, and the economic-aid program created a sudden and disrupting influence in our domestic economy. First came the conversion to a war-effort type of economy. This was followed by gradual, but economically significant, population shifts to urban centers.

By the end of 1941 the United States was at war.

The first year of the decade under consideration was characterized by a sudden transition from a partial war-effort economy to an all-out total conversion to war. Civilian goods, particularly

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durables, became practically nonexistent. All major industry, automotive, textile, chemical, petroleum, and the like, was placing its total productive capacity behind the war effort. Food processing and agriculture produced for the armed services. Skilled trades and unskilled labor were at a premium. Rationing and vital-products allocations became necessary for survival. This meant, obviously, the practical elimination of markets and the distribution process.

During the period, 1942 to 1945, marketing and distribution reached what might be termed an all-time low in terms of twentieth-century development. This period saw the institution of government control over the means of marketing and distribution and the gradual encroachment of government into the lives of the governed. Without considering its eventual effects, we recognize that this was a necessary and important decision and contributed to the war effort.

At the industrial level, many companies simply eliminated their sales staffs (or had them eliminated by the draft) while others kept token forces, mainly for institutional reasons, and to traffic goods to customers. Problems of transportation, problems of advertising, problems of promotion, and problems of research had to await their solution because of the greater effort which had to be made.

A secondary influence, not felt during this period, was the build-up of tremendous liquid assets among the civilian population. Wages, particularly in defense industry, were all out of proportion to the wage scales of the preceding decade. With the elimination of consumer goods and high wages, the creation of fluid assets and intense consumer demand for products became of increasing concern to government economists. Even the plea for investment in government bonds offered at best a stop-gap toward the eventual inflationary cycle which inevitably must occur.

Reconversion—1946-1948

The period from late 1945 through 1947 showed a tremendous conversion effort. With the ending of hostilities, a new war on the domestic marketing front immediately developed. This was the scramble for the consumer dollar. The deep reservoir of liquid assets, the return of the GI from all theaters of war, the rapid increase in family spending units, and the universal demand for housing, appliances, clothing, furniture, and automobiles offered a lush market to tap.

This conversion process grew without recourse to scientific planning and scientific management. It was not necessary to know the depth of consumer demand. Brand identification and brand loyalty were less important than market opportunity. He who was there with a house, a bed, or an automobile found immediate acceptance. This was indeed a period of the "seller's market." Competition was mainly in terms of timing. The salesman of this period—if he can be called a salesman—functioned primarily as an "order taker." Selling and, behind it, the marketing and distributive process were merely a matter of having the greatest amount of goods at a given point with the greatest demand in the soonest possible time. This period allowed little recourse to the scientific understanding of the problems of marketing and distribution. Most economic trends were distorted. Inventories were low because manufacturing could not keep up with the demand. Prices spiraled upward because of customer competition in the market place. Profit margins were high and the costs of nonscientific management's inefficiency in distribution easily could be passed on to the consumer.

This period, however, saw the incubation stage of scientific marketing. The return to civilian production of able men gradually brought back into realignment some of these disruptive processes of 1946 and 1947.

The Return to Normalcy—1948-1950

It was necessary that this situation change radically. The year 1948 showed the beginnings of this change. Consumer demand was being satisfied. The huge reservoir of liquid assets was reaching the normal level, and for the first time since 1939 merchandise, both durable and nondurable, had to be sold. The return to the market of old-established brands, increased adaptability of industrial design, increased efficiency in marketing and distribution, and the applications of scientific management to meet an ever-increasing competitive situation came during this period. This gave rise to the minor recession of 1948. Throughout this period, government became an increasingly important factor in our American economy. While no longer the welfare state of the 1930's, certainly paternalism was the order of the day. The farmer and the labor unions benefited most from a paternalistic government. The small businessman and big businessman found themselves regimented by an increasing burden of control and taxes.

The temporary cutback of 1948 was stimulated, however, by other factors. Many states passed Veterans' Bonus acts. Parity support of farm prices gave another impetus to our domestic economy, and rapid expansion of road building and further industrial capacity brought a halt to this temporary recession and again set an upward trend to prices, wages, and consumer spending.

Korea—1950-1952

As the economic forces found themselves out of balance through the years 1948-1950, the Korean action gave a new and rather important twist to our economic development. We might call this period—"the 20 per cent defense economy." Historically, it is interesting to note a period such as this was developed during the major portions of the nineteenth century in England. The great impetus of the Industrial Revolution came during a period in which England was either in a state of all-out war or was in preparation for additional colonial expansion. A 20 per cent war-effort economy was the order of the day. The physical and psychological effort seemed to increase, not in arithmetic proportion, but rather in geometric ratio to the trend of physical output per man-hour expended.

In the United States, the Korean War saw the institution of government controls, the rapid allocation of strategic materials, and a slightly reduced inventory of consumer durable goods. Yet, the consumer's role, during this period, was much as expected. He went on a spending spree in the expectation of a shortage of civilian goods. His experience during World War II was such as to cause it. Businessmen, acting on a similar impulse, began to accumulate raw materials and stepped-up production schedules. As a consequence, wholesale prices rose approximately 17 per cent and consumer prices, durables and nondurables, rose about 8 per cent between 1950 and 1951. Then inventories again started to pile up and prices had to be cut to move them. Spending, conversely, fell off despite the steady rise in personal income and by mid-1952, savings reached the highest level since the end of World War II.

The Decade—1942-1952

The following major and influential changes characterize the last decade:

1 The redistribution of population between urban and suburban areas.
2 The industrial status of women due to the war effort.
3 The increased length of human life.
4 The range of commodity availability.
5 Increased speed in communication.

6 The redistribution of income.
7 Increased governmental activities.

Comment is necessary on but two of these economic forces as they apply to marketing and distribution. The first is the shift in population.

The increased mobility of our population, with 44 million automotive units on the roads, means that we are literally a "nation on wheels." The latter half of the decade saw a definite shift to the suburbs. Land and housing requirements created the need for "lebensraum." Rapid increases of family units following demobilization saw thousands of developments, moderate in price, offered under the government loan guarantees. Each had to be furnished. Concentrations in suburban areas during the period 1932-1941 ranged from 10 to 40 family units per square mile. By 1950 this had increased to 80 units per square mile.

Following, and dependent upon, this population shift was the rapid rise of the suburban shopping center, the expansion of branch department-store operation, and the rapid increase of the super market. The population being mobile and distance being less important than immediacy, we found an increasing dependence on single-day family shopping. No longer were people going to the corner drug store or grocery store for their necessary items; they found it more convenient to travel greater distances at less frequent intervals to buy greater volumes at self-service markets.

The second great factor as related to marketing and distribution is the equalization of personal income. The great personal fortunes of the nineteenth century had almost disappeared. Tax loads, inheritance and otherwise, had tended to break up this concentration. Labor and the farmer had benefited at the expense of the white-collar worker, proprietor, and professional man. Yet, even the white-collar worker was as well off economically. Extensions of personal credit enabled amortization of personal debt, and the average wage, in current dollar units, gradually increased. Tremendous corporate capital investment aided this redistribution in terms of wage and productivity. The rise of incentive and bonus plans also aided materially in this process. The net result was that more people could buy more goods when they wanted them.

With the end of 1952 these two vital economic forces have pointed the way rather clearly to the decade ahead.

The Decade—1932-1942

Let us look briefly at the decade 1932-1942. This was the period during which the entire efforts of business and government were concerned with pulling our economy out of the depths of the great depression. It was primarily a period of "pump priming" and "make work." The economic waste involved in digging one hole only to fill another placed a tremendous burden on the shoulders of the taxpayer, present and future. It had its effect, however, in keeping a small fund of fluid dollars passing through an abnormally low economy. Unemployment at an all-time high, staggering relief loads, and an exaggerated national fear complex gave instability in terms of demand, marketing, and distribution. The luxuries of scientific management were enjoyed by but few, while the hazards of intensive selling to a practically nonexistent potential were experienced by many. This period came to its logical end with the beginnings of the conflict in Europe, the lend-lease program, and the start of our war economy.

The Decade Ahead—1952-1962

We have looked critically at the period 1942-1952 and have pointed out seven basic influences on the national economy, each of which has a primary influence on marketing and distribution. We have pointed out that the present shift of population from wholly urban to suburban areas had its start following World War II. We have pointed out the rise of the super market, the rise of the suburban shopping center, and the rise of branch department-store operation. We have indicated further that the population shift has attained concentrations approximating 80 family units per square mile. We anticipate that this population shift will continue to increase during the decade ahead. However, it will not be too much of a shift in terms of area, but rather a shift in terms of concentrations, as suburban units tend to gather more families. We anticipate that by 1962 concentrations in the degree of 80 to 160 families per square mile might be characteristic of our standard metropolitan areas.

Since 1945, costs in terms of unit dollars have been increasing at a fantastic rate. The increased tax load has produced gradual, but inevitable, changes in the economy. Manufacturers have had to rely more and more upon scientific methodology in the understanding of their markets and the distributive process. During the nineteenth century, industrial expansion was largely a matter of individual intuition. The second and third decades of the twentieth century then saw a change from individual intuition as the predominant management function to a more decentralized type of management operation. Responsibilities were delegated, product lines were separated from total industrial production, and gradually, following World War II, management saw itself relying more and more upon staff functions in the analysis of its markets, its efficiency of operation, and the demand for product. Market research as a management tool had been a luxury of many large concerns during the period of the 1930's. The success of the public-opinion pollsters, with the quota method of sampling, in predicting election results had given an impetus to an understanding of markets by the test method. During this same period the Department of Agriculture, the Bureau of Labor Statistics, and the Bureau of the Census had been engaged in applying the mathematics of probabilities to the solution of social problems and the mechanics were shown to work. During the last war many department heads and management executives were assigned to the war effort. During this time they came in contact with many of the advances in demographic theory. After their release from government service, these men returned to industry realizing the function of this new device. As a result, market research became a vital factor in supplying management with the answers to consumer demand, consumer attitude, brand preference, and product acceptance. The decade to come will see the development of the scientific principles of social disciplines to a higher and higher degree of proficiency.

The development of time and motion studies as applied to the problems of distribution, particularly in the field of sales efficiency, materials handling, office and clerical work, inventory control, and pricing and unit cost studies will become more scientific and find greater confidence at the management level.

The evidences of the late 1940's point toward a leveling of seasonal fluctuations in nondurable-goods purchasing. No longer are television sets bought at Christmas time, and furniture purchased in the spring of the year. The wide fluctuations of the 1920's and 1930's apparently are leveling out. We anticipate that this trend will continue during the decade ahead. Finally, and of tremendous importance, will be the imposition of self-discipline in industrial management. An understanding of consumer demand and the relationship of capital expenditure to consumer demand will tend to give executive management a clearer insight into the marketing picture. The statistical information supplied by trade associations, the development of economic forecasts, and the function of market research will act as stabilizing factors in the control and production of goods for the future economy.

In all probability, we shall continue with a moderate defense economy through the decade to come. This fact should present a challenge to all engaged in the marketing process.

Personnel Administration

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THE last decade has seen more interest, progress, and development in the management of personnel than in any other previous period.

Wartime Personnel Problems

During the past 10 years personnel managers have been confronted with problems of such infinite variety under pressure of time that all of the techniques previously developed were called into play and many new ones were initiated. The decade started with a period of full employment, with the impact of the war causing rapid expansion in many industries and a corresponding layoff in others. The recruiting of new employees was developed to a high art, as was the balance of the employment techniques. Interviewing, security checks, medical examinations, and so on, were put on a streamlined basis. No sooner were the needs of the work force met than the personnel department had to deal with the problem of the draft. Retention of essential skills or at least a delay in the operation of the Selective Service practices was imperative in war plants if a skilled work force was to meet the requirements of Uncle Sam.

The shortage of manpower caused personnel departments to turn their attention to the employment of women. Jobs were broken down to fit the physical capacities of women and special attention was given to orientation into the factory and training in single skills in order to integrate them quickly into the work force. As more and more women joined the work force, special problems developed. Many companies, recognizing the dual role of women as homemakers and industrial workers, developed facilities for shopping, baby care, and other special services such as beauty parlors, and the like.

A further tightening of the labor market led to employment of many who were physically handicapped. Jobs were engineered so that they could be performed by such people. This is a particularly bright chapter, since the efforts were more than repaid by the performance.

Continuous operation with round-the-clock shifts developed the need for in-plant feeding, and management of cafeterias became an important part of the personnel manager's work.

Rationing and the organization of car pools inevitably found their way into the personnel department.

The training problem was perhaps one of the most important functions of this department. Where large numbers of workers were required for single-purpose operations, vestibule schools were set up in which machinery and equipment, assembly benches, etc., were placed which would duplicate the actual working conditions when the employees became full-fledged operators. Training was on scrap pieces and rejects, and in many plants this salvage operation was a profitable by-product. On-the-job-training became accelerated. Here it was found that many foremen who were quite capable of operating a department had little or no experience in "breaking in" or training new workers. The Training Within Industry operation of the Defense Mobilization Board produced the JIT courses which were designed to give the foremen and supervisors rapid instruction in how to introduce new employees to the job. Many special job instructors were developed who added training to their regular work when required.

A further tightening of the employment market forced the personnel department into the recruitment of employees on a part-time basis.

Reconversion of Industry

With the end of the war came problems arising from the reconversion of industry from a wartime basis to a peacetime basis. Here the personnel management was confronted with problems of layoff and seniority. During the time lag between reconverting from war work to peace work, the added problem of the absorption of the returning veterans required attention. Training programs were developed for the retraining of lost skills. During this period industry was re-engineering work loads and placing more and more operations on a mechanized basis. This also called for special training effort.

Government controls and change in government labor legislation added to the burden of the personnel department. The War Labor Board, with its freeze of wages under the "Little Steel Formula" of 15 per cent, made provision for interplant inequities as relief. Wage and salary departments became quite active in the gathering of comparable wage data in order to establish the basis for petitions to the War Labor Board for increased wages. Today's excellent wage-data techniques stem from such war practices.

The War Labor Board developed its "Maintenance of Membership" formula in return for organized labor's "no strike" pledge. After the war the movement to organize foremen and supervisors into unions developed considerable strength. The Labor Management Act in 1947 retarded this movement, and management began to examine the relationship of the foreman to the balance of management. This resulted in considerable emphasis being placed upon foreman development. The relationship of compensation of foremen to that of workers was examined carefully. Responsibilities and authority were spelled out more adequately. The objectives and concepts of management were clarified and every effort was made to persuade the foreman that he was indeed an important member of the management team.

The aftermath of the war also brought the realization that there were some very real problems connected with the executive personnel. Many executives had worked long hours during the war period and were in poor health from overwork. The expanding volume of goods demanded to meet the needs of the civilian population after the war soon showed that there was a scarcity of competent executive material and that there was virtually no depth to the management team. College recruitment and management-trainee-development programs became a highly important activity of the personnel department. Attention to compensation problems, pensions, group insurance, health examinations, and so on, soon became a part of the comprehensive personnel program with respect to executives.

Changing Concepts and Attitudes of Top Management Toward Personnel Administration

Perhaps the most significant change during the past 10 years has been the change in attitude on the part of top management, which has begun to realize that the personnel policies and programs of the company cannot be made effective unless they are carried into effect by the day-to-day actions and participation of all echelons of management.

Prof. J. D. Brown mirrors the views of the top executives of

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some 84 companies toward personnel management in a recent study which he made. They indicate:

1 That during the past 10 years personnel policies have been given an increasingly important place in the agenda of top management councils.

2 That today there is a wider participation in personnel matters by higher executives who represent the principal divisions of a corporation.

3 That today there is more frequent participation by the chief personnel officer in discussions concerning the impact of general management policies on personnel relations.

4 That there are numerous devices for the formulation of personnel policy, all of which involve both top management and personnel representatives, and that there obviously can be no line of demarcation between those policies to be decided upon strictly by management and those to be resolved by the personnel director alone.

An increasing number of companies is placing the head of the personnel function on the same level with operational vice-presidents. In many companies the personnel officer reports directly to the president. This has been the trend during the past decade.

Prevalence of Personnel Functions Among Companies

The National Industrial Conference Board conducted a survey in 1947 of 3498 companies employing approximately 6,500,000 persons to determine the prevalence of various personnel functions. The returns from the survey indicated that while the functions of personnel management are numerous and vary widely in importance from one firm to the next, nine activities frequently were reported. These nine functions follow in order of frequency of mention: (1) Employment, (2) safety, (3) labor relations, (4) medical, (5) wage and salary, (6) employee information, (7) training, (8) employee benefits, (9) personnel research.

Employment Practices

Today's employment office is a far cry from the dreary old employment office of yesteryear where the men looking for work lined up in front of a window. Applicants often stood in line several hours before they got to the window. It was a dreary business finding a job in those days.

Today's employment office is recognized as being the most important operation in the personnel function. The effectiveness of the working force, as well as its morale, is frequently the result of the use of good selection and placement practices.

The placement manager of today is a specialist who seeks to get the greatest net return for the company from those applying for jobs.

The first step in the process is to determine what the requirements are for the job. Considerable progress has been made in job analysis, which establishes what the worker does, how he does it, the scope of the job and its interrelationship, and the skills involved. One of the major tasks is keeping abreast of changing job requirements.

The use of requisitions, which fixes responsibility on the part of the line operator, is growing and these job analyses become an important part of such requisitions.

The needs of the war produced many new and novel recruitment devices. However, it has become standard practice for the employment department to keep in constant contact with all sources of qualified applicants. The objective is to provide a ratio of recruitments which will vary from 3 to 1 for unskilled jobs to as much as 100 to 1 for highly specialized positions, in other words, a constant backlog. The importance of good public relations through the employment office is apparent in the trends for more attractive offices, rapid handling of applicants, follow-up letters, and visits to trade schools, high schools, colleges, and professional schools. In connection with college recruitment, which frequently is for management-trainee purposes, teams of interviewers often visit many colleges before graduation to interview applicants from graduating classes. Constant touch with placement offices of colleges is a part of the continuing public-relations efforts of good employment departments.

The increased effectiveness of public employment agencies during and after the war is becoming recognized, especially in metropolitan industrial areas. These agencies have developed good information on the availability of applicants for different types of jobs. They also serve the wider function of co-ordinating the demand for workers between areas.

At this point it is important to note that application blanks have been so developed that it is possible for an experienced employment man to get a fairly close appraisal of the applicant before the actual interview. In this connection national and state legislation has been developed patterned after the Federal Fair Employment Practice Committee, which was a war agency, aimed at nondiscrimination in employment and hiring because of race, creed, color, or national origin of the applicant. Eight state laws have been passed since 1945, in New York, New Jersey, Massachusetts, Connecticut, Rhode Island, Washington, New Mexico, and Oregon.

The use of the weighed application blank as a systematized approach to screening applicants is increasing. Interviews today are no longer of the haphazard variety but are patterned in order to provide information, some idea of the motivation of the applicant, his emotional stability, attitude, behavior patterns, and physical fitness.

Another technique in interviewing which is growing in popularity is that of the multiple interview.

The use of aptitude, intelligence, and psychological tests is increasing. One of the difficulties in the use of tests has been that smaller organizations could not afford the costs involved in employing those experienced in the use of industrial psychological tests. Today a number of well-regarded industrial psychologists have developed and thoroughly validated tests which they believe can be applied without danger of misinterpretation on the part of smaller employers. Testing is being used for the selection of new workers, proper placement of workers, screening present workers for special training assignments, choosing workers for transfer to other departments, supplementing promotion procedures, and assisting in the location of potential executive talent. For many years intelligence tests and aptitude tests such as mechanical and clerical tests were all that was available. Since World War I tests designed to measure vocational interests, as well as to measure personality adjustment, have come into play.

The practice of checking references for security purposes during World War II is still being followed with effective results.

The medical checkup, which establishes the kind of employment the applicant is physically able to perform, plays an important part in the over-all evaluation of the employee.

The work of the employment office is not done, however, until the new employee is inducted as a member of the work force. There is a growing trend toward departmental visitations and explanations of the job that the applicant is to perform in relationship to the end product. Increased use is being made of employee handbooks to give information about the company such as pay days, holidays, vacation, sickness allowances, location of cafeterias, restaurants, first-aid stations, lockers, employee activities, and so on, and to advise as to company regulations pertaining to hours, discipline, incentive plans, safety regulations, and other rules and procedures. The induction and follow-up program is for the purpose of welcoming the new employee in a friendly man-

ner and securing his acceptance as a part of the employee group. Continuous follow-up is a part of the procedure, which is now standard practice in the employment office.

The importance of seniority in labor contracts also has required that the employment office keep up to date daily seniority records and be thoroughly experienced in the operation of the layoff and recall procedures, which are today being more and more preplanned. The layoff procedures require the use of good public-relations techniques, which include interviews for the purpose of making explanation of why the layoff is necessary, what seniority rights are established, procedures with respect to unemployment insurance, and the like. Many companies now make provision for the unemployment insurance representative of the government being present when the layoff occurs in order that forms and applications may be made out without delay.

Safety of Employees

Since World War I all echelons of management have accepted the safety of employees as their number-one responsibility. Rapid expansion of the working force with inadequate training caused a considerable increase in accidents, both as to severity and frequency, at the start of the war. The development of good safety practices advanced as workers gained confidence and skill, and those conditions obtained until the reconversion from war work to the production of civilian goods. Accidents increased during this period until once again the process of retraining was completed. Considerable attention was directed in each plant toward making employees safety-conscious and toward developing the habit of working safely. Training programs, plant posters, contests, constant information to employees were all devices which were used.

During the last decade the safety specialist has been added to the staff of the personnel department to assist the line supervision. He is specially trained to anticipate accidents and frequently is given the responsibility for checking machines and mechanical processes before installation. Varying estimates have been made, but the growing practice is for one safety specialist for every 500 to 1000 men, allowing for variation in floor space utilized. He is responsible for complete reports on all accidents and for a continuing analysis of these reports to determine accident trends. He also maintains liaison with the medical department and the employment department to weed out as many as possible of those who have been classified as "accident-prone." Many employees are victims of improper selection and placement in that the applicant's qualifications were not matched with the job requirements. The safety specialist is responsible for co-ordinating this entire program.

One of the developments during the last 10 years has been increasing attention to the control of occupational diseases. In recent years legislation has been enacted in most industrial states providing for the payment of benefits to persons disabled by occupational diseases. The control program usually administered by the doctor or the medical unit generally approaches the problem from the standpoint of substitution of nontoxic or less toxic material and closure or segregation of the problem-producing process, exhaust ventilation, special treatment of the problem material, and personal precaution. In many plants this has now become a specialized job known as that of the industrial hygienist and represents a subdivision of safety work.

An interesting development since the war is the formation of shop committees usually composed of supervisors and stewards or representatives of the union. Great interest has been shown on the part of unions, especially in plants where labor relations are good, in seeing to it that workers follow safety rules, wear protective devices and, in general, co-operate with the plant management in the furthering of the safety program.

Health and Medical Services

Today health programs go beyond mere repair work. There is a growing demand from industry for the application of the principles of preventive measures and control.

The trend to appoint full-time physicians as the chief health officers of industrial plants is increasing. In general, it is estimated that two to three thousand employees will keep one doctor and two nurses busy full time. Part-time physicians are operating on the "standing-order" basis, which today, incidentally, is more strictly observed than ever before.

Medical and health service programs are flexible but usually cover the following functions:

1 Emergency medical care.

2 Continued treatment of industrial accidents or occupational disease.

3 Regular inspection of health and accident hazards in co-operation with the safety department.

4 Periodic examinations of employees and executives to improve and maintain health.

5 Periodic physical examinations of employees exposed to toxic materials on their jobs.

6 Maintenance and analyzing of sickness and injury records.

7 Examination of employees returning to work after absence from illness or injury.

8 Promotion of health education for employees and their families.

9 Co-operation and collaboration with community resources interested in health matters.

10 Development of plans for handling large numbers of seriously injured employees in the event of disaster.

11 Co-operation with all other services in the plant relating to the health of employees such as feeding, recreation, safety, and so on.

In connection with placement of employees, the program usually starts out with a physical examination and laboratory report in which the applicant is placed in one of three medical classifications:

1 Employees who may do any type of factory or office work.

2 Employees who because of physical limitations or handicaps may be placed on specified jobs only, with caution notice to supervisors that they are not to be transferred.

3 Employees who because of temporary or permanent disability cannot meet the physical demands of any job may be deferred or denied employment. Where they are accepted for selective jobs only, supervisors are also cautioned that they are not to be transferred or reassigned.

The list of physical examinations is growing, indicative of the advance of the health and medical services into the field of preventive medicine. Included among these physical examinations are periodic examinations as determined by the hazards of the work, request examinations if related to occupation, transfer examinations in connection with change of job, return-to-work examinations following prolonged absences for sickness or injury, and key examinations conducted annually for the executive and top-management group. (Here the relationship is confidential, and when the management organization is approached on this basis, these examinations become popular.) Also included are eye tests, rehabilitation, medical counseling, syphilis control, physiotherapy, periodic chest x rays, laboratory tests, blood bacteriological studies, blood typing, urine analysis, and so on.

The health and medical services maintain an interest in the nutritional deficiencies of employees and those illnesses that rec-

ords show contribute to absenteeism, such as respiratory diseases, and the like.

During the past 10 years there has been considerable interest in the psychological factors in industrial health services. These psychological functions have become more and more recognized as a part of man's health and hygiene and have resulted in the emergence of psychosomatic medicine. Many industrial managements maintain a comprehensive mental-hygiene program which functions in three major areas:

1 Personnel selection and placement.
2 Employee counseling and adjustment studies.
3 Supervisor development in human-relations management.

Wage and Salary Administration

During the past 10 years a number of influences have been at work which have resulted in increasing the complexity of compensation problems. Twice during this decade the government has instituted wage and salary controls. The War Labor Board and the Salary Stabilization Unit of the Treasury Department froze wages for the duration of the war. The release from these controls, plus the pressure of collective bargaining which produced six rounds of increases, has distorted the relationship between the various compensation units in each company.

The most recent control program was instituted in January, 1951, by the present Wage Stabilization Board and Salary Stabilization Board. During the interim period between controls, a number of large companies, with plant units in many scattered communities, evolved, for easy administration purposes, programs for the payment of the same wages on a company-wide basis regardless of the local labor market area, which, in turn, because of the large number of workers involved in each community, exerted a pressure upon other employers in such an area. The rounds of increases, which were usually in terms of cents per hour, had the effect of increasing the minimum wage for unskilled workers and compressing the wage structure since the higher skilled workers did not receive the same percentage increase as was obtained by the lower skilled workers. Supervisory and executive compensation lagged so far behind that the foremen who were in the first echelon of management in many cases were receiving less compensation than workers, especially if the workers were on incentives. These forces resulted in management's turning its attention to the problem of the relationship in the compensation in all echelons of the organization. Wage and salary administration is a personnel function which resulted from these conditions.

Since remuneration to executives, supervisors, clerical, and operating personnel of business, industrial, public administration, and other organizations covers such a comprehensive range of activities, many companies have centralized this work under a department or unit specially set up to co-ordinate and supervise all of the necessary activities. The amount of salary, wages, bonuses, profit sharing, stock-purchase concessions, retirement benefits, and other forms of financial arrangements with members of an organization's pay roll is a complex problem requiring the assembly and analysis of data from many sources and the development of a sound wage and salary program from the summarized information. Among the factors considered are the following:

1 Skill and competence of personnel in work performance necessitating careful selection of executives and workers.
2 Job study, standardization, analysis, and evaluation.
3 Time and motion study for setting standard tasks.
4 Merit rating to determine compensation for individuals who by superior job performance clearly should receive increased remuneration.
5 Relative financial condition of company as to its ability to pay above average compensation.

The War Labor Board initiated the widespread practice of securing comparative wage data. This has continued in many areas where confidential surveys that are coded greatly assist wage and salary departments in making determination as to the prevailing wage in the localities. Comparison of job descriptions has developed to a point where job titles, often erroneous, are no longer used. The actual job content is the criterion. The wage and salary department concerns itself with the gathering of basic area wage data, job evaluation, and job specifications within the company, the relationship of incentive earnings to base pay, the relationship of the earnings of workers to the several echelons of management, including that of the first line supervision, merit rating, incentive plans for indirect labor, clerks and salesmen, profit sharing, extra financial incentives for supervisors and executives. It also includes the relationship of the executive pay roll to the real salary produced by increased income taxes, bonuses, and the relationship between all of the jobs in the company from the unskilled worker through to the upper echelons of management. It is interesting to note that many companies today have established rate ranges for the upper echelons of management including operating vice-presidents.

A number of considerations have evolved as a general guide for wage and salary-policy determinations. Progressive companies today adopt wage and salary levels which will attract and hold an adequate supply of workers needed and at the same time allow the company to meet competitive costs. This requires locality surveys and cost analyses. A constant study is maintained of the correct differentials between jobs, which requires a continuous research into job evaluation. This forms a guide to collective bargaining. Regularization of hours of work and numbers of employees, together with adjustment of orders to plant capacities, and the like, requires the study of forecasts, budgets, production schedules, and labor audits. Anticipation of possible overtime or extra shifts, of layoffs and reduced hours, spell out the close working relationship of the operating departments with the wage and salary unit.

Measurement and recognition of individual merit as a guide to man classification and promotion raises the question of whether to adopt systematic merit rating or to concentrate on developing superior supervision, or both.

All of these policy matters require the specialization which has resulted in setting up wage and salary units as a staff function in personnel administration.

(*a*) *Job Evaluation.* So much has been written about job evaluation and it has existed so long a time before the present decade that it is unnecessary to detail trends. Labor unions opposed job evaluation for a number of years. It became useful in collective bargaining during the control periods of both World War II under the War Labor Board and at present under the Wage Stabilization Board.

One of the trends has been toward the development of job descriptions, which are useful for three purposes:

1 To determine the value of the job in relationship to other jobs.
2 To provide job specifications for employment purposes.
3 To provide a basis for on-the-job training courses.

(*b*) *Merit Rating.* There has been considerable opposition to merit rating on the part of organized labor. Each collective-bargaining session shows increased pressure for length-of-service increases rather than increases granted by management on the basis of meritorious performance.

Much experience with merit rating has shown certain limitations. The tendency on the part of employees to regard the merit-

rating period as an opportunity for a pure wage increase is the result of poor selling on the part of management. Merit rating is used to determine training needs and as a basis for assisting in employee development. It is a useful device for correcting and improving and, in general, raising the level of performance in day-to-day assignments. Its weakness lies in the reluctance of the employee's superior to "talk over" the merit rating with the employee. Merit rating is useful in the selection of employees for promotion, for transfers, and in some cases for demotions, lay-offs, and discharges. There have been some general limitations owing to the failure to train the line supervision properly in rating. Frequently merit ratings become just opinions or there is a tendency to rate an individual consistently high or low or average depending upon the rater's over-all impressions of the individual. Some raters tend to bunch the ratings together either toward the high or low end of the scale, so that one department shows a "constant error" in evaluation.

The technique of merit rating has improved considerably. Graphic rating scales, multiple-step rating scales, and paired comparisons are some of the devices used to correct the deficiencies. The ranking order system where the rater ranks numerically the employees on the job in question on the basis of over-all performance (or any single trait) is being used. The basic constant error, however, is still present in this form. The forced distribution system was developed during the war in the United States Army, with the rater being required to distribute the employees among a limited number of categories in such a way that a specified per cent is assigned to each category.

Much work remains to be done in connection with merit rating before it will fulfil its potentialities and eliminate grievances which are a constant threat to the morale of the organization.

(c) *Profit Sharing.* Profit sharing for both employees and executive personnel is increasing. It has advantages from the standpoint of developing a more co-operative attitude and, therefore, higher morale. It might be called "incentive management." It is no substitute, however, for wage incentives. It has the disadvantage of causing a drop in morale when there are no profits to share. Profit sharing for employees is either of the cash-distribution type, trust plan, or stock-ownership plan.

(d) *Executive Compensation.* The present inroad of taxes into executive compensation is such that, in order to maintain an executive force with high morale, extra compensation devices are coming into play. These arrangements may be classified as follows:

1 Deferred compensation.

2 Pensions, including the purchase of life insurance and annuities.

3 Profit sharing.

4 Stock bonuses or options.

The deferred compensation usually takes the form of a trust fund as a part of a tax-reimbursement program.

Training and Education

Next to the proper selection and placement of employees, the most important function in personnel administration is the training and development of all employees. Basically a training program concerns itself with the development of skills, knowledge, and attitude. These three factors make up the program, whether it is directed at line employees, supervisors, or executives; whether the people involved are new in an organization or have been employed for many years.

Most companies which have fully integrated training programs break down the activities as follows:

1 Orientation, which deals principally with newly hired personnel.

2 Worker training, which includes all types of job training and general education offered to employees below the supervisory level.

3 Supervisory training dealing with the skills, knowledge, and attitudes necessary for a supervisor, stressing the human-relations aspect of supervision and an understanding of the administrative tools for organizing, planning, controlling, and improving the processes necessary for the success of the enterprise.

4 Executive development, which concerns itself primarily with preparation for policy making, long-range planning, creative thinking and action in fields where decisions are of great and lasting significance to many people.

The most significant development during the last decade has been in the field of supervisory training, and in the past few years since World War II in executive development. Many devices are used in these programs, including individual speech training for the development of personal confidence and articulation, conference leadership training in how to secure decisions and actions from groups, round-table discussions, participation in the development of administrative policies, multiple-management devices including auxiliary board setups, and annual supervisory conferences which are planned and run by and in which supervisors play the leading parts. Regular management or supervisory letters, regular staff meetings, interplant visitations, lectures, training through teaching, role playing, and case studies are other methods for which there seems to be a growing use. Individual work assignments within a department which are not in the direct field of the supervisor's activities or participation in the work of other departments are frequently used for development and growth. The circulation of magazines and the establishment and development of intraplant libraries are all integrated in a well-developed supervisory training program.

Executive-development programs during the past 10 years have tended to utilize college courses. In the last few years some of the larger companies have developed their own courses covering anywhere from 4 to 6 weeks. Some companies plan their courses around individual job rotation. Others plan their courses around auxiliary boards of the multiple-management type made so popular by McCormick & Company. No one single program seems to be the answer to the executive-development problem, but a series of activities to provide full and continuous development is indicated. The activities grow slowly and depend entirely upon the interest and day-to-day participation on the part of the executive.

One of the most interesting activities which recently has been developed is in the field of providing economic education for all employees in an attempt to develop better understanding of America's enterprise system. Small meetings which, after a short presentation, are developed into discussion groups in which everyone participates on a question-and-answer basis, have been extremely successful. Usually the economic presentation is built around the company's business so that it serves the twofold purpose of acquainting the employees with the company's economic problems and, by using the company's problems, serves to illustrate the basic economics of the country.

Employee Information

Keeping the employee informed on all the things which affect the business has been an important trend during the past decade. House organs competently edited and written on a noneditorialized basis are effective. Annual and quarterly reports of the business—and in some cases the stockholders' report itself—are distributed regularly to all employees. Graphic and simplified

presentations written in lay language have proved to be the most effective. Informative bulletins and attractive bulletin boards have taken their place as media for this communication of ideas. Letters to employees and their families, insertions in pay envelopes, informative meetings, the use of internal public-address systems all are being used with more and more frequency.

One important development has been in management's thinking with respect to its relationship to the plant community. Many companies hold annual meetings with the leading citizens of the community to make a report of the company's activities in the community. Open house, when any member of the community may visit the plant, is developing in popularity.

All of these activities indicate a lively awareness on the part of management that a company's public-relations policy has a direct bearing upon the attitude of its employees and the regard in which it is held in the community.

Personnel Research

In order to keep abreast of the many complex forces affecting the company and its personnel, many managements have developed a program of continuous personnel research. Periodic audits are made of all the factors affecting personnel. Attitude surveys are made, usually utilizing the services of some outside, nonpartisan organization to determine just what the employees—and in some cases the people in the community—think of the concern.

In recent years, personnel research also has concerned itself with the problem of backstopping executives within the company and keeping a continuous replacement table and organization charts up to date.

Conclusion

From the foregoing, which is by no means comprehensive, it readily can be seen that personnel administration has assumed an importance which could hardly be envisioned a decade ago. This is but the beginning, for top management is starting to appreciate that, with its new sense of responsibility as trustee for the future of the enterprise 25 to 50 years hence, the relationship of people in an organization is of importance to the future success of the company.

Public Relations

By A. H. FORSTER,[1] LANCASTER, PA.

Origin of Public Relations

The precise origin of public relations, as an organized activity, cannot be documented, but the ancient guilds of Europe were among the first organized groups to speak for business as a unit. They worked to sell the public on the value of guild craftsmanship and quality and developed hallmarks, the forerunners of today's brand names. The main function of guilds, however, was to secure special privileges for business from the ruling governmental powers. The welfare of the public was not considered. In this respect, guild policies differed sharply from modern public-relations practices which are oriented primarily to secure public understanding and acceptance rather than governmental privilege.

One of the earliest recorded uses of the term "public relations" occurred in Thomas Jefferson's seventh address to the Congress on October 27, 1807. In discussing various uses for surplus funds, he said, "...however they shall be disposed of are questions calling for the notice of Congress, unless indeed they shall be superseded by a change in our public relations..." He was, of course, using the term in its purest sense as describing the relations between the government and the public it served.

The term in its modern business sense is reputed to have been used first by Daniel Willard, president of the Baltimore & Ohio Railroad. This claim is made by the man who headed the B & O's Department of Information in the early years of this century when the practice of public relations was getting its first real start. A need for more accurate and complete information about business was responsible for the first efforts in the field.

Among the pioneers, Ivy Ledbetter Lee is generally recognized as the first public-relations consultant. He opened a publicity office in 1903, and in 1906 attempted one of the first industry-wide projects when he became press representative of the Pennsylvania Railroad and the anthracite-coal operators. By training and experience he was a newspaperman—a financial reporter. He served principally as press liaison between the Wall Street financiers of the day and representatives of the press. He made two valued contributions to the concept of public relations. He did more than anyone else to sell business management on the necessity of "humanizing" business operations, and he promulgated the idea that public-relations thinking must take place at the top level of any organization.

Other pioneers included George F. Parker, an early partner of Lee; James Drummond Ellsworth, of the New England Telephone System; J. I. C. Clarke of Standard Oil; and Edward L. Bernays. Bernays also made a significant contribution by hammering the idea that the function belongs at the top policy-making level. Some feel, however, that he hindered public-relations development by cloaking his operations in an aura of mystery, by insisting that PR is a ritual apart from sound business management that only skilled practitioners are capable of handling. Present-day public-relations men are still trying to correct this erroneous idea.

Prior to 1929, PR consisted largely of explaining business to the public—the humanizing process Lee initiated. When the business bubble burst in 1929, the public relations of business suffered a mortal blow from which there has not yet been full recovery. The lean years of the depression and recovery period brought forth a new school of public-relations men. These practitioners conceived PR as a function embracing more than just telling the facts about business. They felt that good PR meant guiding the actions of a business in order to gain public acceptance as well as explaining those actions to the public.

This new concept made only minor headway in the decade from 1932 to 1942. This is indicated by the fact that public relations was not considered a subject of sufficient importance to be included in the Ten-Year Progress Report issued in 1942. Hence public relations actually "came of age" in the decade now under review. World War II was the single most important factor in bringing this about. The techniques of public relations which had been developed on a small scale in the decade prior to 1942, were employed to weld 140,000,000 people into a unified group for the common purpose of winning the war. The government appealed to the public to work in defense plants, to buy war bonds, to give blood, to serve in the armed forces, and to sup-

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port the war effort in every way. Business and industry, largely deprived of regular products to sell to the public, began to use institutional advertising extensively to keep firm names and brand names in the public eye. Response to such advertising made many business leaders aware, for the first time, that the public is as interested in the firm's character as in the economic value of the goods or services it produces.

Coupled with the intensified use of mass-communication media were many social changes also brought about by the war. Millions of service men and women traveled in foreign countries, observed conditions first hand and emerged from the service with a broader point of view and a new interest in social problems. At home there was extensive migration of industrial workers, with a similar change and broadening of viewpoint. It is fair to say that the American people, generally, came out of the war with a greater interest in national and international affairs and with a greater desire to understand the democratic abstractions of equality, liberty, and democracy. While the war served to heighten this interest, it by no means supplied answers to or increased understanding of the problems of society. Indeed, the war aroused interest which helped create many new problems of adjustment for millions of people.

Some observers felt that the conclusion of the war would mean the end of the need for public-relations work. The opposite has proved true. The manifold problems of the immediate postwar period greatly increased the need for sound public relations. Business and industry first had to shift from the manufacture of munitions to the production of peacetime goods. At the same time it was necessary to reabsorb military personnel into the peacetime economy. Furthermore, pent-up consumer demand for goods could not be satisfied as rapidly as the public demanded. Adding still more confusion in this period was the public's gradual awakening to the fact that Russia is an enemy and not an ally. Consequently, the middle period of the decade under review was characterized by the most widespread wave of strikes and labor unrest in the history of our nation.

Faced with this complex situation, business and industrial leaders realized that they could not do without sound public-relations counsel. Nothing demonstrates this more vividly than the rapid increase in the number of professional public-relations consultants who set up office between 1945 and 1950. It is estimated that more than half of such counselors now doing business got started during that period. The establishment of separate public-relations departments within business and industry was likewise stimulated. Approximately four out of five public-relations departments now in existence were set up either during the war or in the 5 years following V-J Day.

The public-relations function is now well recognized by business, industry, trade associations, public organizations, and government. This does not mean, however, that widespread misconceptions no longer exist or that there aren't unethical operators engaged in practices that go under the name of public relations. All things considered, the profession is moving ahead and gaining stature. Evidence of this is the recent adoption of a code of ethics by the Public Relations Society of America, the largest organization in the field.

The modern definition of public relations, as advanced by the editors of "*Public Relations News*," is "The function which evaluates public attitudes, identifies the policies and procedures of an organization with the public interest, and executes a program of action to earn public understanding and acceptance." From this definition it is quite obvious that public relations embraces more than activity in the field of communication. On a par with, or even above communications in importance, is the job of counseling top management in policy and action decisions affecting the public. Actions still speak louder than words.

It is essential that there be no misunderstanding about the true nature of public relations. To clarify the point, here are some brief definitions from leaders in the profession. Edgar M. Queeny of Monsanto Chemical Company, says public relations "seeks to endow a corporation with that which in an individual would be good manners and good morals." The McGraw-Hill Publishing Company states that it "is a way of life—expressing itself every hour in attitudes and actions affecting workers, customers, and the community." The John Price Jones Corporation explains—"We may define public relations as the words and deeds, of an individual or a group, judged by the common concept of sound human conduct." Paul Garrett, of General Motors, says it "is a fundamental attitude of mind, a philosophy of management, which deliberately and with enlightened selfishness places the broad interest of the public first in every decision affecting the operation of the business."

From these few definitions it is apparent that public relations today embraces far more than mere press agentry or even sound press relations. Since it does consist of molding policies and directing practices as well as explaining those policies and practices to the public, the function must be exercised at the highest administrative policy-making level of the business.

The Business "Publics"

The publics with which any business organization must deal may be divided and classified in many different ways. However, the economic publics are, for most firms, the important ones to consider. They include employees, stockholders, customers, suppliers, and neighbors in the communities where the firm operates plants. Guiding the firm in dealing with these groups is the work of public relations.

The first requisite for competent public-relations planning is to become thoroughly acquainted and familiar with these important publics. How this is done is outlined in detail in the discussion of basic techniques which follows.

Basic PR Techniques. No discussion of public relations should minimize the importance of gaging public attitudes and opinions. Abraham Lincoln said—"Public sentiment is everything. With public sentiment nothing can fail; without it, nothing can succeed." This terse statement might well be taken as the keynote of modern public-relations thinking. If the public-relations representative does nothing else, he should keep the operating executives of a business informed as to the trends in public attitudes and opinions. Businessmen generally, because of their intense devotion to and concentration on the job at hand, unconsciously build a wall around their offices through which public attitudes seldom penetrate. This is the "Ivory Tower" so frequently mentioned. Consequently, it is not surprising that policy and action decisions made within these walls all too frequently are at odds with public thinking. If the PR man is on the job, this should not happen.

How does the public-relations man determine public attitudes and opinions? One modern technique is to use attitude surveys. Such surveys are based upon a scientific sampling of opinion and the application of statistical methods in interpreting results. In the last 5 years business has been making increasing use of such scientific studies to determine the attitudes of employees with respect to their jobs, their supervisors, their pay, and working conditions. Surveys also are used to determine the features customers desire in products, the new products they want, their reaction to pricing policies, and similar matters. Opinion studies also are invaluable in determining public attitudes toward a business or organization in a community. In fact, the scientific attitude survey has been employed successfully in gaging the attitudes of all of the publics with which a business must deal,

though less frequently in connection with suppliers than with the other publics.

The comprehensive attitude survey is, however, not the only means of measuring opinion. Spot-checking of attitudes through depth interviews is also an excellent tool. Several leading industrial firms have used this technique with great success in gaging the attitude of two important publics—employees and residents in plant communities. Representatives of the public-relations department sit down with a selected cross section of employees and hold a prolonged nonobjective interview. The employee is allowed to talk about himself and his job, without being inhibited by corrective or admonitory statements on the part of the interviewer. This method is sometimes more helpful in diagnosing the causes of trouble than the formal attitude survey, although it has a somewhat higher percentage of error when results are used to predict attitudes of the whole group. One firm in particular has developed this method with outstanding success in measuring its standing in plant communities. Firm representatives talk to and seek opinions from the leading businessmen of the community, the heads of the Chamber of Commerce and Manufacturers' Association, the presidents of civic clubs, the superintendent of schools and selected teachers, representative clergymen, small businessmen, political leaders, and an assortment of citizens selected at random. The wealth of information thus secured gives a comprehensive picture not only of community thinking but also of the firm's standing in the community.

Another technique, though less penetrating and accurate, is to measure and evaluate public attitudes by holding forums and meetings. This is best-suited for use with employee groups but it has been used successfully by some firms in determining the opinions of stockholders and of citizens in plant communities. Its principal shortcoming is the hesitancy of many people to speak freely when they are a part of a large group.

All of these techniques will yield satisfactory results provided the public-relations practitioner exercises a high degree of objectivity in analyzing and evaluating the information secured. This cannot be overemphasized. The PR man who hears only what he wants to hear and turns a deaf ear to the things the various publics have to say is making no contribution to the firm or organization he represents. It is relatively easy to get people to say nice things about a company or organization, but it is not so easy to get those same people to understand and have genuine affection and respect for the firm. Therefore, if the organization's policies are based upon a false analysis of public attitudes, only grief can result.

Consultation Important. Once public attitudes have been evaluated, it is the responsibility of the PR representative to consult with and advise management so that the firm's policies and practices can be oriented in the public interest. This does not mean that a business must sacrifice its basic objective of earning a profit in deference to public opinion, as some falsely assume. It does mean that when several avenues lead to a specific goal, the avenue which promises the least public resistance should be chosen. If the interests of the business demand a certain decision, despite probable public resistance, it is the duty of the firm, with the help of competent public-relations counsel, to explain the reasons for the decision and to strive to gain public understanding of its necessity.

Obviously, skill in communications is a requirement of success in public relations, and a full knowledge of media is essential. The medium or media most suitable for specific situations must be selected in accordance with the advantages and limitations of each. For example, a firm desiring to keep its employees posted on the status of labor negotiations would not be likely to use the advertising in the public press for that purpose. The general public would be informed through the news columns of the press and perhaps even with paid advertising. However, employees would be given the facts more directly through bulletin-board notices, plant newspapers, letters to their homes, or special meetings addressed by representatives of management.

In the last 10 years, great strides have been made by management in mastering the skills of communication, particularly with employees. However, much further improvement is needed before true management-employee understanding can be achieved. New devices for disseminating information are being devised from day to day. One novel idea recently has been used with success by several large industries. When an especially urgent message must be gotten to management representatives in a manufacturing unit that is spread over a large area, the message is transcribed and placed on the automatic telephone. Supervisory personnel are notified that the message is available by auto-call signal. They may then go to the nearest telephone, dial a set number, and listen to the recorded message. This is but one example of American ingenuity applied to the problem of communication. However, the consensus of public-relations representatives is that face-to-face, word-of-mouth communication is still the most effective for a business or organization to employ in dealing with its publics. Recognition of this marks a milestone in PR. It is a far cry from the day, not much more than a decade ago, when management largely dealt with its publics by cold, informal printed communiqués issued from the Ivory Tower.

Dealing With Specific Publics. The most interesting developments in public relations of the decade under review are techniques designed for use with specific publics that have enjoyed marked success. One of the most important areas for PR attention is employee relations. Many firms consider this function important enough to be handled by a separate department, either outside the PR department or as a subdivision of it. Even when the employee-relations function is outside the framework of the formal PR organization, there is usually top-policy PR co-ordination.

The broad viewpoint of public-relations men has done much to force management recognition of the fact that sound personnel policies, alert and intelligent supervision, and good working conditions are factors that generate high morale, reduce employee turnover, cut accident losses, increase productivity, minimize work stoppages and labor-management friction.

The communication skills of the public-relations operatives have been equally helpful in promoting a two-way flow of information between management and employees. Employee magazines, plant newspapers, and similar forms of printed periodicals have increased in number and improved in quality of editorial content under the guidance and sponsorship of PR representatives. Similarly, PR has guided management in doing a more comprehensive and effective job of reporting and interpreting the financial results of operations to employees through the use of illustrated and simplified annual reports, slide films, and motion pictures.

Many firms today are holding employee meetings at regular intervals in which mutual problems are discussed in a friendly, open manner. In some instances these meetings are conducted by representatives of top management, in others by the foreman or supervisor to whom the employees directly report. There can be no doubt but that this communication device has made significant progress toward eliminating misunderstandings and building a sound basis for employee-management co-operation.

One development of the last 5 years deserves special comment—the intensive effort of management to give employees at all levels thorough training in the basic economics of business operation, and to increase their understanding and appreciation of the prin-

ciples underlying the American form of representative democracy. No matter how the actual training may be handled, whether by an industrial-relations department, foremen, or top-management representatives, this economic education effort has primarily been the result of public-relations thinking. Not all economic education has been sound by any means, and some efforts probably have done more harm than good. On balance, however, progress is being made. Opinion surveys are verifying the fact that a sound course in economics will interest employees and that it will do much to generate understanding of profits, productivity, the role of capital, competition, the dangers of government intervention, and how our economic system shares.

One of the newest mediums of communication pioneered by PR in the field of employee relations is the information rack. Its basic purpose is to stimulate employee interest in economics and the social sciences generally, and to broaden their reading horizons. The rack service operates on a voluntary basis. A wide variety of publications, ranging from treatises on economics, philosophy, and political science to discussions of fishing and household hints, is supplied in racks placed at convenient points within the plant or office. Employees take any booklet that interests them and the pickup has been remarkable, ranging occasionally as high as 100 per cent of employment. The information rack is only 3 years old but has already been accepted and used by more than 1000 firms.

The task of dealing with the customer public is basically the responsibility of the merchandising departments of the business—selling, advertising, and promotion. However, in businesses recognizing the PR function, there is always top-policy co-ordination by PR representatives. The extent of the public-relations influence in merchandising varies, of course, with the type of business. Firms engaged in the manufacture and distribution of consumer goods have a larger public-relations task than firms engaged in the manufacture of capital goods, although both types of businesses must have selling policies based upon sound principles of fair dealing. Public relations perhaps did its biggest job in the field of customer relations during the immediate postwar period when goods were in short supply and systems of allocations had to be set up to insure fair distribution.

Supplier relations give public-relations men the least amount of trouble. If the basic policy of dealing with firms that supply raw materials, parts or subassemblies is sound, little remains to be done except to follow through within the organizational framework to see that purchasing representatives adhere to the policy. This does not mean, however, that this phase of public relations is not exceedingly important. A firm becomes known as a decent, honorable organization or as a chiseler rather quickly on the basis of its dealings with suppliers.

Relations with stockholders have come to the fore in recent years. In the decade from 1932 to 1942, and for the first 5 years of the decade under review, the stockholder was the forgotten man of American business. The pendulum of management attention seemed to have swung completely in the direction of employees and away from stockholders. For this reason PR men have been giving increased attention to building good relations with the owners of the business in order to stimulate investment. One of management's greatest concerns today is that unfavorable tax laws are causing the capital market to dry up. Public-relations men have attacked the problem of generating a greater stockholder interest in the business by giving the financial statements a face lifting and by preparing slide films and motion pictures to explain the report for showing at regional stockholder meetings. Special attention also has been given to making annual meetings of stockholders significant through careful program planning and the use of detailed charts and analyses to give the owners a more complete picture of the state of the business.

In addition, there has been a marked increase in the number of stockholder publications. These are usually issued quarterly and are designed to keep the owners of the business informed of the interesting and significant facts of operation during the periods between stockholder meetings.

The task of community relations is one that has become almost exclusively the responsiblity of public-relations personnel. Until PR men called management's attention to the problem, many firms made no planned and organized effort to discharge their responsibilities as good citizens in the communities where they operate plants. Such efforts as had been put forth were generally haphazard.

It is impossible within the space limitations of this paper to do justice to the many projects that have been carried on by business and industry in recent years to improve community relations. Only a few of the more significant projects can be covered. These include special communications in the form of newspapers, news letters, or informal personal letters from the firm to community "thought leaders" which include business and professional people, clergymen, teachers, and public officials. Such communications usually are chatty in nature and strive to explain the problems of the business and to demonstrate the firm's concern with community affairs. Open houses and guided tours of business and industrial establishments are not new in themselves but have been given new significance in recent years by making them more than mere inspections of operations. Talks by management personnel, informal discussions, and presentations explaining the economic significance of the operation have been added. Many firms have further specialized the tour idea by conducting visits for particular groups. Some of these groups include school teachers, school children, doctors, dentists, lawyers, and clergymen.

Representatives of business are now taking a more active part in community affairs, serving on boards of hospitals and Community Chests, holding nonpaying public offices, working on committees, serving as leaders for youth groups, sponsoring athletic teams in recreational projects, and working closely with school authorities through parent-teacher organizations. Supplementing this work, sound public-relations policy calls for business to carry its full share of the cost of maintaining the community's health, character building, and charitable institutions, such as hospitals, Community Chest, and Red Cross.

Being a good neighbor in the community also calls for the elimination of industrial nuisances, such as smoke, dust, or stream pollution. A new awareness of this responsibility has come about largely through the top-level pressure of public-relations representatives.

These are but a few of the more significant developments in community relations of the decade. The alert, progressive, public-relations-minded firm finds countless ways to demonstrate its sense of community responsibility. One outstanding example will serve to illustrate this. When an explosion virtually leveled a southwestern city, a company operating a plant there chartered airplanes and flew in doctors, nurses, blood plasma, and other medical supplies. Top executives of the firm were on hand to assist personally in the organization of disaster relief work. Employees of the company who were victims of the explosion were searched out and given personal assistance. This was one of the outstanding jobs of public relations of the decade, yet it involved not a single written word for public consumption. The actions in the public interest spoke much more eloquently.

The Future of Public Relations

Looking to the future, it is logical to expect that public relations should achieve complete recognition as a top-management function within the next 10 years. New techniques will be de-

veloped for measuring and evaluating public opinion more precisely. One such development is now under way. It is the scale-analysis method of interpreting survey results. Essentially it is a scientific method of determining where each respondent to a questionnaire stands on the scale between the poles of liking or disliking a firm. Scale analysis also assists in eliminating errors of interpretation.

Further improvement also is to be expected in the use of the media of communication, with particular emphasis upon the oral form. Management representatives, following the leadership of PR personnel, may be expected to become more skillful in talking with the various publics of business on a face-to-face, give-and-take basis.

And finally, a forecast based upon a trend that already is beginning to be evident, more and more public-relations representatives will find their way into top-management positions. In 1951 alone, the issues of *Public Relations News* reported that 41 PR executives had been elevated to the level of vice-president with complete responsibility and authority for handling the PR function of their organizations. Four were reported as having been upped to executive vice-presidents. More than 20 were named to serve on their company's board of directors. These records are not complete but they do indicate an upward trend of public relations to the top policy-making bracket where it belongs. Indeed, it is essential for the continued success of our business system that this be so, for we have entered an era in which the relations of a business with its publics will determine more than any other factor whether it is able to operate continuously and earn satisfactory profits.

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Labor Relations

By GIDEON M. VARGA,[1] NEW YORK, N. Y.

The happy announcement of V-day was the beginning of a series of new industrial-relations problems. The widespread cancellation of war contracts, the curtailment of overtime, and the resulting smaller take-home pay created an epidemic of unrest and strikes. The dilemma was increased further by the re-employment of veterans and operation under discrimination laws passed in several states. The "no-strike" war-duration agreement that existed between some managements and labor groups and which was instrumental in developing a closer relationship, was forgotten quickly upon the enactment of the Taft-Hartley Act which succeeded the Wagner Act. The Wagner Act had governed labor relations since 1935. Organized labor denounced the Taft-Hartley Act which was passed over the president's veto by a better than 4 to 1 vote in the House and 3 to 1 in the Senate, as a "slave-labor law."

Strikes Costly

More than 500,000,000 man-days of work have been lost since World War II and over $5,000,000,000 in wages was lost by workers. The latest labor news release issued in November, 1952, reported 3,200,000 man-days of idleness in September, 1952, as a result of work stoppages. This represented 50 per cent more than in August, and 700,000 man-days more than in September, 1951. It is less by 400,000 man-days than the September, 1947–1949, average of 3,590,000. The loss was not confined to the industry or to the workers in which the strike occurred. It extended into thousands of companies which are dependent upon the supplying industry.

The 1952 steel strike beginning June 2, and lasting for 55 days, was called the "worst production stoppage in American history" by Defense Production Administrator Henry H. Fowler and disrupted industries dependent upon steel. The impact of the strike was predicted to extend well into April, 1953. The supply of iron ore brought from the Mesabi Range is limited by the 7-month lake shipping season and each nonactive mining day results in a half-day ore-supply loss in 1953. While the steel strike was in progress, many substantial layoffs occurred. The industries pinched by the strike were the automobile, freight car, toy, electrical products, and the can manufacturers. Other industries are dependent upon the ones mentioned, such as the agricultural industry. The shortage or the lack of tin cans causes a loss of the perishable fruit and vegetable crop.

The cause for strikes during the 1942–1952 period was not always the monetary issue. The major issues of the three big steel strikes since World War II were:

1946 Wages—an increase of $18^1/_2$ cents per hour.
1947 Pensions. 1952 Union Shop.

In the 1952 steel-industry strike, the companies offered a wage increase of 16 cents per hr, one week after the strike began and again after the Supreme Court upset President Truman's seizure of the mills. The steelworkers' union was willing to accept the wage offer but held out for the "union shop." This issue is extraneous to wages and no gain in wages resulted from the additional 48 days of striking after the wage offer.

The Forgotten Man—The Foreman

The expansion of the working force and the longer hours of work caused increased hardships upon the front-line supervisor, the foreman. This was also due to the meager supply of competent supervisors to meet the influx of employees. As the

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employees' take-home pay approached and in some instances exceeded the foreman's pay, the foreman looked around for means of ameliorating his own take-home pay. In many organizations he was considered an individual and received separate pay increases. The prestige he enjoyed as a member of the management team was lost since his duties and responsibilities on hiring, firing, and discipline changed. The change was caused by the greater utilization of the grievance committee and the shop steward. A grievance procedure was followed which invariably made the foreman, working under pressure and under strained human relations, a defendant in the grievance. The grievance was handled by the wage administrator, industrial administrator, or the vice-president in charge of production. Coupled with this was the wage freeze imposed upon management which prevented giving increases to foremen. Foremen organized their own union and signed a separate contract with management.

The National Labor Relations Board then ruled that under the Wagner Act foremen were prohibited from organizing a bargaining unit. Confusion developed by the interpretation of the loosely worded Section 9B of the Wagner Act. This section established standards for determining the collective-bargaining eligibility of any given group. The confusion was increased further by the failure of the National Labor Relations Board and the War Labor Board to follow a consistent labor-relations policy nationally and also by reversing positions of the disputes in many cases. The employers were not compelled to recognize foremen as a bargaining unit by law. The NLRB also ruled that foremen could not be discharged for union organization activities.

Many companies strengthened their supervisory organization by inaugurating intensive training programs. Through specially prepared training material and bulletins, human relations, leadership qualities, and economics were emphasized.

The sections of Standard Practice Instruction manuals affecting the foreman's supervisory responsibilities were reviewed carefully at foremen's meetings held during working hours. Suggestions for revising and modifying sections were encouraged. A rotating committee of foremen was delegated to assist at and become part of the company's contract negotiating committee. The intensification of the foreman's responsibilities made him a part of management instead of the "forgotten man."

Re-Employment of Veterans

Many controversies developed as veterans were discharged from the armed services and returned to their jobs in industry. Some of these clashes were between the veterans' personnel division of Selective Service System and the unions. Practically every company had one or two veteran re-employment problems which reached the arbitration stage of the grievance procedure. The principal cause of these controversies was the GI Bill of Rights, passed in June, 1944. This bill does not levy specific obligations upon the employer as was recognized under the earlier 1940 law which limited coverage to 1 year of military service. Management was confronted by the problem of rehiring veterans when hundreds of faithful wartime employees were discharged because of contract cancellations. In many instances the veteran bettered himself while in the service and refused to accept his old position. The multiplicity of veteran cases kept the labor-relation managers occupied to an even greater extent when they were confronted by the bill's provision that veterans could not be discharged from their positions without cause within a year after their re-employment.

Labor-Relations Stabilization

Recognition of the detrimental and costly effect of strikes by both companies and unions led to attempts to stabilize labor relations in the United States. The General Electric Company made an extensive study to determine the influencing factors in industrial relations. The General Motors Company attempted to stabilize labor relations by negotiating and signing a 5-year contract. To offset any increases in living costs over the 5-year period, an escalator clause was made part of the contract. The escalator clause is used to adjust wages either up or down. The wages adjustment is based upon the government's cost-of-living index as published by the Bureau of Labor Statistics. The contract was instrumental in preventing strikes.

At the time the General Motors 5-year escalator type of contract was signed, it was considered a great advancement in labor relations. Unfortunately, the Bureau of Labor Statistics, unaware of General Motors' and the unions' intention to use the index as a wage adjustment or regulator, had started a 3-year study to revise the Consumers' Price Index in 1949. The inflationary spiral, the Korean war, and the rush by other companies to follow General Motors in establishing escalator clauses in their labor contracts placed the Bureau in an embarrassing position. To extricate itself from this situation the Bureau issued an interim index, into which were incorporated some of the preliminary revisions. Upon the simultaneous publication of the old and interim indexes, union economists raised objections to the interim index. The interim index lags slightly behind the old index and the price changes are depressed. The change in buying habits has lower food weight.

In 1952 The Aluminum Company of America signed a contract similar to the General Motors contract. This contract is subject to the approval by the Wage Stabilization Board and depends upon the Office of Price Stabilization for granting the company the request for a price relief. Generally, fewer long-term contracts of this nature are being signed.

Unions and companies with escalator clauses face difficult problems when the new 1953 Bureau of Labor Statistics cost-of-living index, based on 1947–1949, is issued. This makes obsolete the "old" and the "interim" indexes. Lack of complete understanding of wage computations lead to distrust. Both the unions and management will be faced with the problem of understanding how to convert from the old base of 1935–1939 to the new base despite the Bureau's conversion table.

Owing to the lower change of rate in the new index, compared to a corresponding percentage change in living costs, under the previous index modifications in the escalator clauses are imperative to provide the same wage increases as obtained under the current index. Many arbitration cases are anticipated. Statisticians will be called upon to act as arbitrators to settle wage controversies.

Labor Legislation and Government's Influence on Collective Bargaining

During the war and immediately following the war the Government was instrumental in the development of industry-wide bargaining. This is summarized capably in a paper presented by Dr. E. H. vanDelden as follows:

1 The Government via the War Labor Board during wartime, established commissions, arbitrators, and referees on a national and area industry basis.

2 The National Labor Relations Board, by its decisions determined that multiple-unit bargaining and industry-wide bargaining conditions provided a basis for sound collective bargaining units.

3 The Fair Labor Standards Act and the Walsh-Healey Act have established industry-wide minimum for wages.

4 The OPA established price adjustments on an industry-wide basis.

5 The War Production Board rationed materials on an industry-wide basis.

6 The War Manpower Commission and the Selective Service developed manpower controls and administered the draft on an industry-wide basis.

The rapid growth of industry-wide bargaining since the war is dramatic proof of the existence of such influences.

The Fair Labor Standards Act (Wage-Hour Act) was the last major labor law until World War II. The protection offered by the National Labor Relations Act was instrumental in increasing the union membership. As the unions increased their strength both economically and numerically, they also increased the apprehension of employer groups and several public groups, since the unions were wielding considerable political power. President Franklin D. Roosevelt was idolized by labor for 12 years. President Truman at the beginning of his term had labor support. The employer group joined by public groups protested against the uncurbed union practices and powers.

Since the National War Labor Board, established in 1942, by executive order, lacked authority to settle wartime labor disputes, Congress enacted the War Labor Disputes Act in 1943. This empowered the government board to regulate labor-management relations.

As the feeling increased that "labor" was accumulating more powers than was considered good for the public welfare, Congress passed a series of laws to curb certain union activities. The Lea Act, the Hobbs Anti-Racketeering Act, the "portal to portal" law and finally the controversial Taft-Hartley law in 1947 were written.

The Taft-Hartley Act changed many provisions of the National Labor Relations Act. It listed unfair practices for both companies and for unions. Many controversies developed, and the unions vigorously opposed and resented the Act. The provisions of this act and other labor legislation are listed in the sections on legislation passed during the 1942–1952 period (see Appendix).

The minimum wage was raised to 75 cents per hr. In many industries this was inconsequential since only the lower class or beginners were receiving less than 75 cents per hr. The majority of workers were well over the legal minimum wage.

In July, 1952, a new Wage Stabilization Board was created. Its members consisted of representatives of labor, industry, and labor. The old board's power was more extensive. It was noted that only those labor disputes which endangered the defense program were referred to the board. In those cases where the White House did not refer the case to the Wage Stabilization Board the unions were permitted to strike. The Wage Stabilization Board would intervene only in those disputes which became serious.

The new board was not looked upon with favor by the unions since its function is limited to wage and salary disputes. It cannot make nonwage recommendations nor terminate disputes or strikes, nor can it recommend the union shop. This will prevent the recurrence of strikes similar to the 1952 steel strike.

The organization of employees in many companies was instrumental in including in the bargaining agreement a grievance procedure which invariably included the use of arbitration for settling disputes. With the increase of such agreements it became natural that more cases were taken to arbitration during the 10-year period from 1942–1952, than in previous periods. The settlements were more readily accepted and the use of arbitration was advanced, since both management and labor referred disputes to arbitration only as the final step.

Prediction for Next Ten-Year Period

In the realm of labor relations, speculation on future developments and trends becomes a tremendous task. Reflecting upon the outcome of the 1952 presidential election, the fear that labor had become politically powerful may be dispelled temporarily. The victory of the political party which was held responsible for passing the Taft-Hartley Act, despite labor's first historical public announcement of supporting the opposing party's candidate, may result in a more intensive effort by labor if the president elect does not support labor and keep his election promises.

There is a strong probability that the new Congress will introduce a law to ban industry-wide bargaining by limiting union officials to negotiate labor contracts to only one company. It is felt that limiting the bargaining at the company level will compel an independent solution in each company. Attempts to create laws unfavorable to labor may result in a wave of protest strikes.

Another possibility in the way of new labor legislation may be the abolishment of the National Labor Relations Board and replacing all or some of the present five members and the general counsel.

As an aid to settling labor disputes, President Truman urged the chairman of the Federal Trade Commission to institute a study of the breakdown of the consumer's retail dollar into labor, materials, distribution costs, and profits. The facts will establish the effects of wage increases on labor costs, profits, and prices. Once such data are published the use of the cost-of-living index as a wage-adjustment measure may be abandoned.

The refinement of such studies coupled with investigations and development of man dealing with his fellow man will not result in a Utopia but will advance industrial relations to the benefit of all.

Appendix

The pertinent labor legislation which had a bearing on labor relations is made a part of this report to provide a means of comparing legislation with previous ten-year periods and future ten-year periods.

Public Law No. 552[2] approved 5/13/42. Amends section 1 (c) of the Walsh-Healey Act to exempt from the overtime provisions employees who are exempt under the Fair Labor Standards Act because they are employed on an annual basis for more than 2,080 hours in the pursuance of a bona fide collective bargaining agreement.

Public Law No. 729, approved 10/2/42. Authorizes the President on or before November 1, 1942, to issue a general order stabilizing prices, wages and salaries on the basis of levels existing on September 15, 1942. Empowers him to make adjustments with respect to prices, wages and salaries to the extent that he finds it necessary to aid in prosecution of the war or to correct gross inequities.

Prohibits action taken with respect to wages and salaries

1 Which are inconsistent with the provision of the Fair Labor Standards Act or the National Labor Relations Act, or—

2 For the purpose of reducing wages or salaries below the highest paid between January 1, 1942, and September 15, 1942.

Gives the President power by regulation to limit or prohibit the payment of double time except when, because of emergency conditions, an employee is required to work for 7 consecutive days in any regularly scheduled work week. Penalty of $1000 or 1 year imprisonment or both for violation of the act may be imposed.

Public Law No. 34, approved 4/11/43. As a rider to the Public Debt Act of 1943. Amends the Wage Stabilization Law (Public Law No. 729) to delete the following clause: "Provided that the President may, without regard to the limitation contained in clause (2) adjust wages or salaries to the extent that he finds necessary in any case to correct gross inequities and also aid in the effective prosecution of the war.

[2] Annual Digest of State and Federal Labor Legislation.

Public Law No. 89 (passed over the veto of the President 6/29/43), War Labor Disputes Act. Amends Section 9 of the Selective Training and Service Act. Empowers the President to take possession of plants, extends to any plant, mine, or facility where suspension of operation due to a labor disturbance interferes with the war prosecution. Requires return to private operation of any business taken over within 60 days after restoration of productive efficiency. Prohibits taking possession after the war ends and terminates authority to operate any plant taken over 6 months after the war. Plants taken to be operated under the terms and conditions of employment in effect at time of possession. Provides that the government agency operating the plant or a majority of the employees may apply to the War Labor Board for a change in wages or other conditions of employment. Makes it unlawful to coerce or encourage any person to interfere with operation of a plant in possession of Government by a lock-out, strike, or slow down or to aid any such lock-out, strike, or slow down by giving guidance in its conduct or by providing funds for the conduct thereof, or for the payment of strike, unemployment, or other benefits to participants.

Gives the War Labor Board statutory authority and extends its powers by giving the Board the right, whenever a labor dispute may lead to substantial interference with the war, to initiate hearings. Provides that failure of either party to appear at such a hearing does not deprive the Board of jurisdiction to proceed to a hearing and order. Provides that the Board in deciding a dispute and ordering changes in wages, hours, or other conditions of employment must conform to the provisions of the Fair Labor Standards Act, the National Labor Relations Act, the Price Control Act and all the other applicable laws.

Empowers the Board to issue subpoenas compelling attendance of witnesses and production of records and to apply to any Federal Court for an order requiring persons to obey subpoenas. Prohibits Board member from participating in decisions in which he has a direct interest as an officer or representative of either party. Provides that the enumerated powers of the Board are not applicable to any matter within the purview of the Railway Labor Act. War production plants are required to file a notice of intention to strike with the Secretary of Labor, the War Labor Board, and the National Labor Relation Board. Makes it unlawful to strike until a strike vote has been conducted by the National Labor Relations Board on the 30th day after notice has been filed. Persons who violate this section are liable for damages to any person injured thereby or the United States.

Public Law No. 135, approved 7/12/43. Title IV specified that no part of the funds appropriated for the National Labor Relations Board shall be used in connection with a complaint case arising over an agreement between management and labor which has been in existence for 3 months or longer without a complaint being filed.

Public Law No. 373, approved 6/28/44. Title IV. Limits the Board's use of finds in certain types of cases but does not amend the National Labor Relations Act. The NLRB may challenge the validity of contracts entered into with allegedly company-dominated unions. (The former "rider" prohibited this action unless a charge was filed within 3 months from the execution date of the contract.) The NLRB can proceed upon unfair labor practice charges filed by an employee or employees of the plant affected where contracts have been renewed, even though no changes are made in the terms of the contract, if such charges are filed within 3 months after the date of renewal.

Public Law No. 49—"Portal-to-Portal Act" (approved 5/14/47). Claims arising prior to May 14, 1947. Relieves an employer from liability under the Fair Labor Standards Act, the Walsh-Healey Act, or the Davis-Bacon Act because of his failure to pay an employee minimum wages or overtime compensation for any activity engaged in prior to the date of enactment (May 14, 1947), except an activity compensable by express provision of a written or nonwritten contract or by custom or practice not inconsistent with such a contract in effect at the time of activity at the employees' place of employment.

Provides further that, with respect to employee actions brought under the Fair Labor Standards Act on future wage claims, the court may "in its own discretion" award less than the "additional equal amount," as liquidated damages as previously required, or none at all where the employer shows that he acted in good faith and had "reasonable grounds" for believing that he was not violating the act. Amends Fair Labor Standards Act so as to bar bringing employee suits under that statute by agents or representatives designated by employees but permits collective or class suits brought by employees in behalf of themselves and other employees.

Public Law 101 (passed over veto 6/23/47). Labor-Management Relations Act of 1947 amends the National Labor Relations Act. Creates an independent agency for conciliation of labor disputes. Restricts certain boycotts, union activities, and unlawful combinations and establishes a joint congressional committe to study and report on basic labor problems. Major provisions are as indicated in the following. Amends the NLRA to remove from the coverage of the act, supervisors and independent contractors. Enlarges the NLRB from 3 to 5 members and vests final authority for investigation and prosecution of all unfair labor practice cases and authority over all field personnel in the Board's General Counsel to be appointed by the President. Provides that employees have the right to refrain from union organization except to the extent that such right may be affected by a valid union-security agreement authorized under the Act.

Makes numerous changes in the provisions of the National Labor Relations Act with regard to unfair labor practices. Prohibits exercise by an employee of obligations under a "closed-shop" agreement. Permits "union-shop" agreements only with a union which is the representative of the majority of employees in a unit appropriate for collective bargaining and only if a majority of all eligible employees have voted in a Board election to authorize the union to make such an agreement. No union-security agreement is authorized if in violation of a state law. Provides that no employer may discriminate against an employee for nonmembership in a union if he has reason to believe that membership was not available on the same terms generally applicable to other members, or if the employee lost membership for any reason other than the failure to pay dues or initiation fees.

Lists six unfair labor practices of labor organizations or their agents, including: (1) Restraining or coercing (*a*) employees in the exercise of their rights to self-organization, to engage in concerted activities and to refrain from any such activities, except as the latter may be affected by a valid union-security agreement authorized pursuant to the Act, and (*b*) employers in the selection of their representatives for collective bargaining or the adjustment of grievances; (2) causing or attempting to cause an employer to discriminate against an employee on the basis of a union-security agreement not permitted under the Act or against an employee who has been denied membership on grounds other than the failure to pay dues; (3) refusing to bargain collectively; (4) conducting or encouraging a strike or a concerted refusal to handle or work on materials if the object is (*a*) a boycott, (*b*) furtherance of a jurisdictional dispute, (*c*) forcing any employer to recognize a union if another union has been certified by the Board, (*d*) forcing some other employer to recognize a union which has not been certified by the Board, (*e*) sympathy strike, (*f*) reassignment of work tasks, or (*g*) requiring any employer or self-employed person to join a labor or employer organization; (5) requiring employees covered by a valid union-security agreement

as a condition precedent to becoming a member, to pay fees which the Board finds excessive or discriminating; and (6) forcing or requiring any employer to pay any money "in the nature of an exaction for services which are not performed or not to be performed." Also makes unlawful activities constituting unfair labor practices under foregoing and gives persons injured thereby the right to sue the union and recover damages.

Specifies that expressions of any views or arguments shall not constitute evidence of an unfair labor practice if they contain no threat of reprisal or force or promise of benefit. Defines "collective bargaining" as the obligation of the parties to meet and confer in good faith and to execute an agreement, but does not obligate either party to agree to any proposal or to make concessions or to discuss changes in a contract for a fixed period to take effect before the end of such period unless the contract provides for such reopening.

Specifies that neither party may terminate or modify an agreement without giving 60 days' notice to the other party, with maintenance of existing terms and conditions until the date of such termination or modification. Provides that an employee striking within 60-day period loses his status under the Act unless rehired by the employer. Requires the party desiring to terminate the agreement to notify the appropriate federal and state mediation agency within 30 days after the original notice if no agreement has been reached by that time. Authorizes employers to petitition for a certification election after a union has claimed a majority and demanded exclusive recognition. Permits employees to petition for an election to obtain representation for collective bargaining or to determine whether an existing collective bargaining representative shall continue to be the representative of the employees.

Provides that as a prerequisite for petitioning for Bid of Elections or investigations (1) the union must file annual copies of its constitution, by-laws including financial reports with the Secretary of Labor and furnish copies of financial reports to all its members. (2) That each union officer must file an annual affidavit stating that he is not a member of or affiliated with the communist party and does not believe in or belong to or support any organization that believes in or teaches the overthrow of the Government by force or unconstitutienal methods.

Amends the National Labor Relations Act regarding the prevention of unfair labor practices.

Directs the regional attorneys of the Board to apply for injunctions against strikes or concerted refusals of employees to use, process, or handle goods, or to perform services if an object thereof is an unfair labor practice involving (1) a boycott, (2) forcing recognition of a union other than that certified, (3) reassignment of work tasks (if relief is appropriate), or (4) requiring an employer or self-employed person to join a union and if, after investigation, the regional attorney or other officer has reasonable cause to believe that the unfair labor practice charge is true.

Establishes a Federal Mediation and Conciliation Service independent of the Department of Labor. Requires this agency to mediate either on its own motion or at the request of one or more of the parties. Authorizes the Service to assist the parties in using other means of settlement, including submission to the employees of the employers' last offer for acceptance or rejection in a secret ballot. Directs the Service not to mediate disputes which have only a minor effect on commerce, if state or other conciliation services are available. Makes it the duty of employers and employees to (1) exert efforts to make and maintain agreements, (2) confer on request of the other party to a dispute, and (3) participipate in such meetings as may be called by the Mediation Service. Establishes a 12-member national labor management advisory panel to the Mediation Service.

Sets up special procedure for disputes affecting an entire or substantial part of any industry which imperils national health or safety. Authorizes the President in such cases, in the event of actual or threatened strikes or lock-outs, to appoint a Board of Inquiry to investigate and report on the issues and to direct the Attorney General to petition a federal district court for an injunction. Lifts the restrictions of the Norris-LaGuardia Act on the courts, for this purpose. Requires the Board of Inquiry to make a further report to the President within 60 days after issuance of the injunction and during this period the parties are also required to confer with the Mediation Service. Directs the NLRB within the next 15 days to hold an election on the question of whether the employees involved in the dispute wish to accept the final offer of settlement made by the employer. Directs the Attorney General within 5 days thereafter to request that the injunction be vacated. Requires the President to submit a full report to Congress of the proceedings together with such recommendations as he sees fit if the dispute has not been settled.

Public Law No. 393, approved 10/26/49; effective 1/25/50. Amends the Fair Labor Standards Act of 1938. Increases the minimum wage from 40 cents to 75 cents per hr. Substitutes in the definition of the term "produce," the phrase, "or in any closely related process or occupation directly essential to the production thereof," for the phase, "or in any process or occupation necessary to the production thereof." Revises the overtime-pay provisions to clarify what payments to an employee must be included and what payments may be excluded in computing the required time-and-one-half compensation for hours worked in excess of 40 in the workweek.

Modifies certain minimum wage and overtime exemption; adds complete minimum wage and overtime exemption for certain additional classes of employees. Modifies the child-labor provisions.

Public Law No. 734. Social Security Act Amendments of 1950 make important changes in the Old Age and Survivors Insurance program and in the federal grants-in-aid programs of public assistance and welfare services. It extends coverage of self-employed individuals, farm laborers, domestic workers, most Federal civilian employees, etc.

Public Law 914, approved 1/10/51. Amends the Railway Labor Act to permit carriers and employees covered by the act to bargain collectively for the union shop and check-off. The Railway Labor Act, as amended in 1934, had expressly prohibited provisions for the union shop and check-off in collective bargaining agreements in industries covered by the Act.

Public Law 914 permits carriers and labor organizations to make agreements requiring, as a condition of continued employment, that all employees shall become members of the labor organization representing their craft or class, within 60 days following the beginning of employment or the effective date of the contract whichever is later. Specifies that agreements may provide for check-off of periodic dues, initiation fees, and assessments, provided the employee gives the employer a written assignment of such dues, fees, and assessment. Written assignment shall be revocable in writing after the expiration of 1 year, or upon termination of the collective bargaining agreement, whichever is sooner.

Public Law No. 51, approved 11/19/51. Extends for 4 years the authority to induct men in the armed forces, to terminate on 7/1/55, and in effect, extends present operation of the Selective Service Act of 1948 for the same period. Makes changes in reemployment provision of the 1948 Act, so that one or more enlistments after June 24, 1948, confers re-employment rights of total service thereunder, does not exceed 4 years, unless extended by law. Sets forth the conditions of eligibility for statutory re-employment rights.

Public Law No. 189, approved 10/22/51. Amends the Labor Management Relations Act of 1947, to eliminate the requirement

for a special election conducted by the NLRB before a union shop provision may be included in a collective bargaining contract.

Public Law No. 429, approved 6/30/52. Defense Production Act of 1950 amended in 1952. Provides for price and wage stabilization; provides for the settlement of labor disputes; establishes a system of priorities and allocations for material, and by these measures to facilitate the goods for national security and for other purposes.

Public Law 590, approved 7/18/52. Social Security Act Amendments of 1952. Increases old age and survivors' insurance benefits.

Cost Accounting

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COST accounting during the past decade has been dominated by war, inflation, and governmental regulation. The first half of the decade was covered by World War II, while the latter half has been given over to "preparedness," a state which we still have with us. Inflation has been present for the entire period, of course, as a result of deficit financing and the large proportion of the national production devoted to war and defense requirements. Government controls of essential materials and defense profits affected practically every business to some extent and were a major problem for most.

Another and perhaps corollary influence has been the reaction of organized labor to the combination of controls, a favorable political atmosphere, and inflationary pressures. In an effort to increase real wages under stabilization regulations, round after round of demands for wage increases were accompanied by demands for so-called "fringe" benefits.

Business was under continued pressure to get out production first to meet war needs, then later to fill war-deferred consumer demands. While prices were regulated and taxes took away most of the profits, margins before taxes were generally good and there was a slacking of pressures to establish close control of costs and expenses.

At the same time, some constructive forces were at work. One of these was continued interest in improved techniques of management. Business and technical associations in particular helped to develop and spread better practices. Special attention was given to better organization and the problem of executive incentives under near confiscatory taxes.

Under these influences, cost accounting inevitably has undergone a number of changes. Some are merely reactions to abnormal war conditions, others are reactions to continued progress, but for purposes of discussion they can be grouped as follows:

1 Adaptations to government contract and control regulations.

2 Reactions to inflation.

3 New requirements for labor negotiations.

4 Further developments in controls and in management needs for information.

The problems of accounting for government contracts were already present at the beginning of the decade, having started in the late 1930's as defense preparation. For important segments of the national economy, war production, either under prime or secondary contracts, became the principal, if not the only, occupation. Accounting for such work called for greater detail and actual costs. Standard costs in some cases were not acceptable. Hence there was a renewed interest in actual costing methods.

In other cases it became necessary to increase the number of variation accounts to obtain the required actual costs.

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In still others there was a return to job-costing techniques, methods which had been discarded in favor of the greater control values of process costing.

The government, through tax and procurement legislation, defined what expenses were admissible and what were nonadmissible. Accountants perforce followed these classifications and discarded, wherever they were inconsistent, the older terminology of direct and indirect labor and materials. Greater breakdown was essential, and every charge had to be documented for audit.

Government Controls

To the special requirements of government contracts were added those of regulation. Profits were "renegotiated" or subjected to "excess-profits taxes." Prices were "controlled." Materials were allocated under a series of regulatory plans, culminating in the Controlled Materials Plan.

While renegotiation affected only those companies engaged directly or indirectly in war production, all business felt the effects of other controls. Each control created its own need for voluminous records and new details. The pressure of government accounting requirements forced the transfer of personnel from more normal activities to the task of making out the government reports. Since the demand was to "get out the production regardless of cost," control activities went by the board. The only constructive analysis and planning in addition to operations and financing were those called for by the need for new facilities.

However, not all regulations were unfavorable to business. During the war, and again in preparation for defense against communism, some provision was made for accelerated amortization of fixed assets. "Certificates of necessity" were a prerequisite, but where they were available it was possible to depreciate assets, for tax purposes, over a 5-year period. Multishift full operation increased wear and tear and became an acceptable basis for accelerated depreciation, even without "certificates of necessity." Many companies charged on their accounts higher depreciation than was allowable as a tax deduction.

There was a steady rise of prices during the war, although price controls were in force the entire period. Shortly after hostilities were over, the combination of relaxed controls, pent-up demand, and deficit financing let loose inflationary pressures, and prices rose even faster. When to all this were added the new requirements of preparations for defense against communism, the price increases became spectacular, particularly in the case of basic materials.

Inventory gains, and risks, and the problem of replacing fixed assets on the new price levels, caused businessmen great concern. There was a renewed interest in the Lifo plan of inventory valuation. Many companies which had failed to take any action on the plan earlier, now adopted it. Techniques to extend it to retail inventories were developed and the necessary governmental approvals were obtained to permit their use.

However, no satisfactory accounting approach was developed to provide for the replacement of fixed assets at higher prices. The only solution continued to be a financial treatment, the reservation of profits to provide the necessary additional capital. When profits were insufficient, companies either had to seek new outside capital or not replace at all.

Fringe Benefits

The decade under review introduced into the cost-accounting picture to an extent never before known, the concept of fringe expenses. These included an almost infinite variety of pension costs, insurance coverages for employees, overtime and shift premiums, incentive pay, bonus pay, vacation and holiday pay, and other new costs arising from negotiated labor agreements.

Efforts at wage stabilization did not prevent direct increases in wage rates. What they did was put difficulties in the way of such direct increases, and promote demands for other fringe benefits. Faced by the combination of a shortage in the labor force and a political atmosphere friendly to organized labor, business could not refuse such demands. Hence, in negotiating the most favorable contracts under the circumstances, and in trying to recover the resulting costs, management of necessity turned to cost accountants for assistance.

Up to this time the practice largely had been to handle such minor fringe expenses as were incurred, through indirect or overhead expense techniques. The growing importance of these costs made that impossible. New bases and techniques of distribution were necessary. One result has been an enormous increase in the clerical costs of preparing the pay roll.

In general, the last half of the decade has been characterized by high-volume production. Had it not been for this, the profit picture would have been far different, for costs have risen almost continuously. This includes indirect, particularly clerical, costs. Moreover, inflation has pushed up prices, while labor and government have claimed, and obtained, a larger share of the sales dollar.

At the same time, however, lack of materials has interfered with production. Prices have been regulated. Inevitably, breakeven points and margins have suffered so that only volume has been favorable. As the decade closes then, even volumes higher than ever before are not enough to prevent general and substantial declines in net income after taxes—which is all that is of any value to the shareholder.

It is natural that under these conditions there should be a renewed interest in management controls. Thus the control of direct material and direct labor has advanced considerably. Averages of past performance are no longer accepted as good enough. Also, intensive production planning and production control, detailed engineering studies of material utilization, labor utilization, equipment utilization, and methods of manufacturing have had considerable influence upon cost accounting for control purposes.

The result is that cost accounting, at one time limited to the manufacturing processes, has been extended to cover warehousing and receiving; finished stock, shipping, and delivery; and, to a more limited extent, selling and sales promotion, as well as administrative, purchasing, and the various clerical functions, including the cost department itself.

In the cost department today there is a trend toward less detail on costing for inventory and cost of sales, with an increase in cost detail for control purposes. Many companies use a standard cost for a finished product, charging finished stock and crediting the manufacturing expense variation account at the end of each month. This applies equally to job-cost systems and process-cost systems. Admittedly, some variances occur in the inventory in process each month, but these usually are not serious and are corrected easily. No detail of actual dollar costs is kept by jobs or processes, though a notation is made of date, department or operation, and time. This record is used in combination with the production control and scheduling work, although the actual cost cards are in the cost department.

Except where absolutely essential, direct labor and other direct or indirect payments or benefits to employees are treated as manufacturing expenses in the same manner as indirect labor or supplier costs. The items, however, are broken down into properly described items small enough to be controlled easily by the responsible supervisor.

The budgeted cost or the standard cost, calculated at a departmental or operations rate, is used as a yardstick to control and measure the actual labor, supplies, and expenses used by a department or on an operation.

A tremendous amount of detail clerical work has been avoided at very little if any loss of real values and a considerable increase in control information. As previously mentioned, this type of work has extended into the sales and administrative functions of the business.

It is not new for cost accountants to recognize the importance of responsibility in management and controls. Accounts and budgets increasingly have reflected individual responsibility. Thus, when businessmen extended the use of decentralized management, cost accountants found it relatively simple to adapt their accounts and budgets to show the new responsibilities.

When businessmen went even further and used profits as an incentive and measure of performance, this too caused little trouble. Cost and profit statements by divisions and plants became common during the decade. Cost men took yet another step and allocated assets on the same basis in order to build a base against which gains could be measured to determine relative profitability. This has constituted real progress.

Reducing Costs

One final trend deserves comment. The pressure upon margins has forced a new interest in selling and administrative expenses as being relatively fertile ground for cost reductions. Warehousing, delivery, selling, and clerical costs are being brought under the microscope. The feeling has developed that these costs can and must be reduced.

This has led to the increasing application of techniques that are successful in analyzing and reducing manufacturing costs, to selling and administrative costs. Large insurance companies, in particular, set out to develop work standards for clerical operations, standards which could serve as a basis for more efficient planning, scheduling, and better controls. While there was nothing really new about this, except in its introduction to a different area, it did offer great promise of substantial cost reductions.

Perhaps the most important consequence of the past 10 years in the cost-accounting field is a new type of cost accountant who apparently is far too busy to write about his profession.

This cost accountant is thoroughly familar with government regulations affecting labor, material controls, and price stabilization. He has become well grounded in sales and administrative policies, procedures, and problems. He has a first-hand acquaintance with budgets and budget preparation. Because he works with and performs part of the work formerly associated with production planning and control, he has become completely conversant with the plant and with pay-roll procedures.

In short, during the last decade the cost accountant has come of age and can take his place beside the professional accountant, the engineer, and the lawyer with no feeling of inferiority.

It is hoped that in the next decade he will find time to tell about it in well-prepared, authoritative, instructive books detailing the newer techniques.

Federal Administrative Management

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THE past 10 years have been as momentous in the field of governmental management as in industrial management—possibly more so, because it has been upon government that the major responsibility has rested for mobilizing the entire country to carry through a war to successful conclusion, to guide postwar adjustments, and to establish policies and programs for waging peace. A special degree of inventiveness, flexibility, and sustained leadership has been required of federal management.

The organizational and administrative achievements in the conduct of World War II represent the work of American organizational genius at its best. The administrative problems involved in liquidation of most of the special government structure created for World War II presented, in some respects, a more difficult problem than the initial action to create the civilian war structure. It is slightly more difficult to make eggs from an omelet than it is to make an omelet from the eggs!

After World War II we did not repeat our 1919 mistake of crawling back into our shells of isolation, making no effort to keep the rest of the world from flaring out and embroiling us again in a world conflict. We began before the troops of the enemy had surrendered to create the United Nations and other international agencies, to organize the United States Government for participation in these collective enterprises to keep the peace, and to carry out a variety of foreign programs aimed at building a stable and peaceful world.

But the Soviet Union was not content to let these programs, including the remarkably successful work of the Economic Cooperation Administration in the reconstruction of the Republic of Korea, go forward unmolested. The North Korean communists, under the guiding hand of the Kremlin and later with the active support of the Chinese communists, started a "hot war." Aggressive hostilities were already flaring in Indo-China, and militant subversive operations were being conducted elsewhere. These conditions presented the fourth major challenge of the decade to federal administration—the necessity for the establishment of a vast peacetime defense program, calling anew for a variety of emergency civilian agencies within the Executive Branch and for strengthening our military institutions.

In addition to the successful handling of the special problems of two mobilizations, one demobilization, and organizing to meet new international commitments, the decade saw significant improvements in the permanent organizational structure of the Federal Government, a marked increase in the effectiveness of the Presidency in executive management, and notable improvement in the internal management of federal departments and agencies.

World War II

By 1942 many of the main outlines of the federal organization required for the conduct of war had become clear. It had proved necessary in pre-Pearl Harbor days, and for some time after we were actually in war, to rely upon relatively improvised arrangements pending public support for emergency powers enabling the President to meet head-on the administrative adjustments necessary under the vast new burdens falling upon the Executive Branch.

After the declaration of war, it was necessary to transfer these agencies into operating organisms which could drive forward with all the vigor necessary to deal with the increased requirements for materials and trained manpower and for the production of weapons and other military and civilian essentials. Effective controls over the use of resources had to be established; programs for a vast expansion in production launched; methods of handling manpower and labor problems strengthened; civilian defense measures instituted; and overseas economic operations expanded. (Basic Lend-Lease and Bureau of Economic Warfare programs had begun in 1941.)

It was natural that the Government concentrated on getting such operating agencies functioning under a full head of steam before centering its attention on mechanisms of co-ordination. Accordingly, it was not until October, 1942, that the Office of Economic Stabilization was established to co-ordinate the programs and work of agencies such as the Office of Price Administration, War Labor Board, War Manpower Commission, War Production Board, and War Food Administration so far as their activities affected economic stability. In November, 1944, the Office of War Mobilization and Reconversion was established as the final over-all instrument to reconcile the competing demands for resources of the different agencies and to resolve issues which could be settled short of direct intervention by the President, who was already overburdened.

Comparison With World War I

Contrasted with its performance in World War I, the government's administrative achievement during World War II was extraordinary, reflecting a tremendous advance during the two intervening decades in the capacity of the federal bureaucracy to apply sound organization and management principles to new situations. Achievements in the military services were equally significant. Certainly one of the most important organizational developments of the second World War was the March, 1942, reorganization of the War Department and the subsequent establishment of the Joint Chiefs of Staff.

In scientific-management terms, perhaps the most important administrative difference in the two wars resulted from the 1942–1945 work of a central management staff under the President. Reorganization of the Bureau of the Budget by its new director, Harold D. Smith, in 1939, and the extension of its functions, had equipped it to provide central planning, review, and facilitating service essential for the orderly development of government structure and programs in peacetime as well as in times of emergency. Consequently, on December 7, 1941, a cadre of organization and management experts was already at the disposal of the President. Many of these persons had studied United States experience in World War I and were well informed on the administrative arrangements of other countries preceding us into the war. During the war and through the period of postwar adjustment, this group of experts identified urgent problems, drafted executive orders or legislation to meet them, helped set up the new agencies, advised departments with regard to better administration of their functions, worked out delegations of authority and assignment of functions, and generally facilitated co-ordination and effective operation. The Bureau's Division of Administrative Management, which carried the brunt of this work, became a pattern for emulation throughout the world.

Just as a management planning and consulting staff was an essential for the Government as a whole, the new emergency agencies, as well as old-time departments, found it essential (as

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does any large private agency) to establish such facilities. The most striking developments along these lines were the activation of control divisions in the Army Service Forces and the establishment of various administrative management, management engineering, and organization and planning units in the Navy, the Army Air Forces, and the Coast Guard. These offices were charged with working out and installing improved organizational arrangements, programs of work simplification and work measurement; with improving procedures; reducing reports and paper work; and effecting other changes aimed at efficient and effective operations.

The establishment of similar administrative management units in most of the war agencies likewise contributed to their ability to adjust to their changing burdens and to work out the myriad administrative problems that beset them.

Civil Service During the War

In the vast personnel problem faced by the Government early in World War II the question arose as to whether Civil Service should be stored in the attic for the "duration" or whether it was possible to adjust its rigidities and cumbersomeness to meet the fast-moving demands of mobilization. The decision was for "adjusting" and a variety of emergency arrangements were made to decentralize personnel functions to the agencies. The Civil Service Commission put major emphasis upon assisting agencies with recruitment of personnel and other problems rather than upon imposing routine restrictions. Yet, at the same time, attention was focused upon standards of selection and ways of eliminating persons who failed to measure up.

The success of this approach to personnel problems has carried on to the present day, so that increasingly the personnel operations of the Government are conducted on a constructive basis in lieu of the negative, restrictive approach which inevitably characterized early Civil Service reform.

Many of the wartime gains have carried over into the permanent service:

1 Agencies have been delegated authority to take final action on many types of appointments, promotions, and transfers which previously required the Commission's approval. In taking these actions agencies are guided by written standards and instructions issued by the Commission. Periodically the Commission's Inspection Service reviews agency operations on a sampling basis to verify the correctness of actions taken and to make suggestions for improving the personnel program.

2 By the greater use of boards of examiners, agencies are given a direct voice in the hiring process. More than half of all new civil-service appointments are now made by examining boards, trained and supervised by the Commission, but located in the agencies.

3 Far greater authority has been delegated to the Commission's 14 regional offices so that prompter service can be given agencies in the field service.

4 New examining methods for higher-grade jobs give greater stress to the *quality* of a candidate's experience rather than the mere length of his exposure.

To complete the summary of Civil Service developments during the decade it should be noted that greater emphasis has been placed on the recruitment of promising junior personnel. Hundreds of the outstanding members of college graduating classes now enter the service each year in scientific, professional, and administrative posts.

Likewise, progress is being made in career-development plans to insure that competent employees are identified and given a chance by training and planned experience to prepare themselves for higher responsibilities. Many "internship" training programs are operated for juniors both in scientific and administrative fields. At a higher level departments are establishing "executive-development" programs to provide future replacements for key administrators.

Democracy Versus Totalitarianism

All in all, the experience of World War II provided a striking demonstration that administration is more efficient when real democratic conditions exist than under a totalitarian regime. The process of carving out the government's policies, programs, and administrative arrangements appeared disorderly—and it was. But a study of administrative measures taken in Germany under Hitler clearly shows that our organization for handling of allocation of materials, production, and supply problems, and many other war operations was, as a whole, far more efficient. Disagreement, unreconciled differences of view, gaps in operations, and inefficiency flourished under all of the dictators.

War History

When war came in Europe, and the United States started looking towards its defenses, the Bureau of the Budget began to review the experience of World War I. The record that it found was regrettably inadequate. Many official documents had been removed from Washington, a number taken by officials of the day for their personal records. (This practice is now prohibited.)

In order to make sure that the administrative experience of World War II was not similarly lost, the President instructed the Bureau of the Budget to work out a comprehensive program for recording and evaluating experience for subsequent study and guidance. While some of the reports prepared by war agencies under this program are not very useful, others are good. The Committee on Records of War Administration, which supervised the program in the Bureau of the Budget, issued a comprehensive and useful summary study entitled "The United States at War."[2]

Demobilization and Liquidation

In September, 1944, the President directed the Bureau of the Budget to develop plans for the liquidation of the war agencies and reconversion of the Government to peace. Controls were relaxed greatly after V-E day, and the liquidation of the emergency agencies was speeded, in some cases prematurely, after V-J day. By the end of 1945, seventeen agencies had been terminated. The process, however, was far more orderly than in 1918.

A principal reason for the smoothness of transition was the high degree of teamwork achieved between the Bureau of the Budget and the Office of War Mobilization and Reconversion during this period.

The principle was followed of transferring the residual activities and personnel of emergency agencies to the permanent agencies to which their functions were most closely related. Thus the final tasks in the priorities and production field were placed in the Department of Commerce, the wind-up of foreign information and Lend-Lease programs in State, and various manpower activities in the Labor Department.

There were some unfortunate casualties, among them many of the general management staff units, established in the War and Navy Departments during the war. These are now being reinstated or re-emphasized under the impetus of a new emergency. It is unfortunate that the value of organized management staffs is fully recognized only in times of stress.

[2] For sale by the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C. (555 pages). Price $1.50 (paper cover).

Defense Mobilization Again

When the aggressive actions of the Soviet satellites made clear to the world that peace would be violated at any time that suited the Kremlin, the job of mobilization began all over again. The freshness and applicability of World War II experience together with the establishment in July, 1947, of the National Security Resources Board to engage in mobilization planning has enabled the Government to take this new administrative burden in stride.

New agencies have been required in cases where a totally new function or activity had to be performed, as, for example, price and wage controls, civil defense, and so on. Fewer independent agencies have been necessary, however, than during the war. The regular departments have greater ability today than they had in 1941 to carry on emergency functions, and, of course, the crisis is not as acute.

The keystone of the civilian defense machinery is the Office of Defense Mobilization, established in December, 1950, as one of the constituent units in the Executive Office of the President. This office provides general leadership and direction over all of the agencies responsible for defense-mobilization activities. These include old line departments such as Agriculture and Interior as well as new agencies such as the Defense Production Administration, the Federal Civil Defense Administration, the Office of Price Stabilization, and others.

As stated by the Director of Defense Mobilization, a major purpose is "to avoid the creation of a new and massive bureaucracy but instead to rely, as far as possible upon the brains, know-how, and facilities of existing departments and agencies of the Government. Thus problems of fiscal and credit control, of industry, minerals, agriculture, etc.—as they pertain to defense mobilization—are retained in the departments and agencies already set up for these purposes."

Relations With International Agencies

With the signing of the United Nations Charter, the United States assumed a heavy responsibility for multilateral action. Wartime participation in the Combined Resources and Production Board, the Combined Materials Board, and other joint operations helped to equip us for these new international commitments, but by and large, we entered them with very little prior experience to go on.

In the midst of the war the State Department set up a special planning staff under Leo Pasvolsky to develop proposals for postwar international machinery. It was through that staff that United States proposals at Dumbarton Oaks were formulated and our participation in the San Francisco Conference made effective. This unit has evolved into a Bureau of United Nations Affairs under an Assistant Secretary and is divided into offices concerned with economic and social affairs, political and security affairs, etc. It is this Bureau which now provides "backstopping" for United States participation in international conferences and organizations, and co-ordinates the interests of and roles performed by the various federal departments in international agencies.

The Bureau of United Nations Affairs is also responsible for seeing that international agreements are translated into appropriate domestic action, both legislatively and administratively. This latter task is, in many instances, the most difficult of all.

The co-ordinating job is a large one since most of the actual work in connection with the international agencies must be performed, not by the State Department, but by the departments of the Government having a special interest in the subject matter. Indeed we find offices concerned with international problems attached to nearly every federal agency. The Department of Agriculture, for example, through its Office of Foreign Agricultural Relations, handles a major part of the United States activities concerned with the United Nations Food and Agricultural Organization. It also provides assistance to the Mutual Security Agency and the Technical Cooperation Administration of the State Department on overseas agricultural projects.

Of interest in connection with these co-ordinating efforts, is Yale Professor Walter Sharp's study, prepared under the auspices of the International Institute of Administrative Sciences with funds made available by UNESCO, describing the machinery which governments have established to work through international organizations. This study shows that effective international action depends largely upon the competence of national governments in providing effective backstopping to their delegations and in carrying out agreements reached in the international bodies. In all such matters it is necessary for a government to mobilize the assistance of their ministries in dealing with problems falling within their respective spheres. Most foreign offices tend to monopolize these matters without cutting their various technical ministries adequately into the process.

The United States has also had to make special organizational arrangements to participate in various regional bodies within the framework of the United Nations. These include the North Atlantic Treaty Organization, the Organization for European Economic Cooperation, and the Organization of American States.

Administration of Economic and Technical-Assistance Programs

Drawing all foreign military, economic, and technical-assistance operations together under the Mutual Security Program in 1951, was a significant step in the correlation of United States foreign-assistance programs and the administrative arrangements for them. A continuity in United States foreign operations has existed from the establishment of Lend-Lease and the Board of Economic Warfare in 1941, through UNRRA and Post-UNRRA relief, early exchange activities with Latin America, the Truman Doctrine (which provided military and economic assistance to Greece and Turkey), the Marshall Plan, Point IV, the Mutual Defense Assistance Program, and finally the Mutual Security Program. The successive arrangements for conducting these programs are heartening testimony to the government's capacity to build upon and profit from experience.

The Mutual Security Program includes the military assistance to NATO and other eligible countries administered by the Defense Department; "defense support" to European countries and economic and technical assistance to six countries in Southeast Asia and the dependent overseas territories of European countries, administered by the Mutual Security Agency (successor to the Economic Cooperation Administration); technical and economic assistance in South Asia, Latin America, and the Middle East administered by the Technical Cooperation Administration of the State Department; and the U. S. contribution to such U. N. programs as technical assistance, Palestine refugees, Korean reconstruction, etc.

Congress attempted to assure effective co-ordination of all of these parts of the Mutual Security Program by establishing a Director for Mutual Security in the Executive Office of the President. This has helped, but the provision of the law that the director serve also as head of the Mutual Security Agency—one of the constituent agencies which he is charged with co-ordinating—has been confusing.

Similar co-ordinating measures have been taken in the field where the ambassador is charged with providing leadership and general direction over all United States activities in the country and with molding the MSA Mission, the Military Assistance Advisory Group, the Embassy, and any other special representation into an effective working team. The "U. S. Special Repre-

sentative in Europe," as representative of the President and agent of the Director for Mutual Security, the Mutual Security Agency, the Department of State, and the Department of Defense, has been charged with giving general supervision to the functions of these agencies in Europe.

While an increase of co-ordination has been effected, an undue number of independent agencies still handle segments of interrelated overseas technical and economic activities. These include the Mutual Security Agency, the Technical Cooperation Administration of the State Department, the Export-Import Bank, some areas of the Defense Materials Procurement Administration, the Department of Commerce, and the Treasury. Surveys conducted by Mr. Gordon Gray, the Hoover Commission, the Technical Assistance Advisory Board under Nelson Rockefeller, the Committee on the Present Danger, and the Brookings Institute have recommended that foreign economic and technical-assistance activities be brought together, at least in some degree, under a single agency. Rationalizing this structure will be an important task confronting the new Administration.

United States Interest in Administrative Improvement in Other Countries

In another sense also, management improvement has stepped into the international sphere. The United States has found that the ability of other countries to participate effectively in the Marshall Plan and other co-operative economic and technical-development activities and to become effective partners in the NATO and the UN depends in large measure upon the quality of organization and management of their public service. For this reason the TCA, the MSA, and the United Nations have made improvement in public administration an important feature of their technical-assistance programs.

In Greece, for example, the Mutual Security Agency has, at the request of the Greek Government, assigned a considerable number of experts to help improve its budget, civil service, tax, agriculture, health, public works, and other ministries. More modest programs of this type are going forward in Thailand, Formosa, the Philippines, and other countries in Southeast Asia.

In addition, MSA brings officials from these countries to the United States for training in scientific management under the general direction of the Bureau of the Budget. The TCA has an intensive program of this type going on in Latin America, and that program is now spreading to other countries in which TCA operates.

The United Nations, which set up a training center in public administration over 3 years ago, is providing technical assistance in a number of forms to countries wishing to improve their governmental administration. It sends out survey teams, assigns administrative experts, offers fellowships, and conducts seminars on special problems of public management.

Improvements in Executive Structure

This brings us to the simplification in the structure of the Executive Branch made during the decade. While the Bureau of the Budget sought to take advantage of the demobilization period as an opportunity for further rationalization of the permanent governmental structure, it was not until the appointment of the Commission on Organization of the Executive Branch (known as the Hoover Commission) that the stage was set for a period of real accomplishment. The President had found it almost impossible to secure support for reorganization measures during the 80th Congress. Experience shows that President and both Houses of the Congress must be of the same political complexion if action on administrative matters is to be achieved.

Indeed the general disposition of the Congress—to concentrate on policy and program or delve into detailed management problems—has a decided bearing on the success with which the executive can function. The decade's principal retrogression has been that of Congressional encroachment in the executive field. Personnel riders, overstaffed and competing Congressional Committees, one-year authorizations, "witch-hunting" have been serious impediments to effective management.

Most significant change in the government structure, taken in several steps, was the establishment of the Department of Defense and its subdivision into the Departments of the Army, the Navy, and the Air Forces. As part of this change the National Security Council and the National Security Resources Committee were established, and the Joint Chiefs of Staff were set up on a permanent basis.

The consolidation of the Federal Public Housing Authority, Federal Home Loan Bank System, the Federal Savings and Loan Corporation, and the Federal Housing Administration during the war in a temporary national housing agency was finally made permanent, after much Congressional debate on various bills, through the establishment of the Housing and Home Finance Agency.

Repeated efforts to have the Federal Security Agency established as a department were rejected by the Congress. The Federal Works Agency was abolished, and one of its principal functions, the Public Roads Administration, was transferred to the Department of Commerce. With the addition of the Roads responsibility and Maritime Commission functions, the Department of Commerce began to emerge as the central transportation agency of the Government. The other main function of the Federal Works Agency, namely, building construction and maintenance, was combined with the central procurement functions of the Treasury Department, the surplus disposal job, archival and records management, and management of space in general government buildings in the field into the General Services Administration. The establishment of this service had been projected as a result of the Bureau of the Budget studies in the early 1940's, but it was not until 1950 that it was possible to secure Congressional support for the reorganization plan submitted by the President in 1949. The Labor Department was strengthened by the transfer of the U. S. Employment Service from the Federal Security Agency, the Bureau of Employees Compensation, and enforcement of labor standards on federal financed construction, but weakened by the Taft-Hartley Act.

In addition to changes of this sort and the rearrangements of the military services, important reorganizations of the Department of State and Reconstruction Finance Corporation have taken place. The principal feature of the adjustments in the RFC which were highlighted in the public press as a result of questions in regard to the integrity of some of its operations was the substitution of an administrator for a board so far as the general administration and operation of the agency are concerned.

The effect of the major readjustments in the organization of the Executive Branch is reflected in the accompanying chart. Prominent agencies shown on this chart, not found in a chart of 1944, include, in addition to the foregoing changes, the National Science Foundation, the Subversive Activities Control Board, and the National Security Training Commission. Temporary agencies identified with dotted lines parallel a variety of emergency war agencies established during World War II such as the War Production Board, the War Manpower Commission, the Office of Price Administration, the Office of Civilian Defense, the Office of Defense Transportation, the War Shipping Administration, the Smaller War Plants Corporation, and other bodies.

Executive Leadership and Co-Ordination

During the past 10 years the concept of the President as General

Manager of the Executive Branch has taken clear form and much improvement has been made in the general management of the whole executive establishment.

During this decade it has become clear that the establishment of the Executive Office of the President in 1939, with the Bureau of the Budget, shifted from the Treasury Department as a principal component, has represented the most important single contribution to the organization of the Federal Government. With increased acceptance of the idea that the Executive Office should consist of those staff facilities which assist the President in carrying out his over-all responsibility for leadership, management, and co-ordination of the Executive Branch, a number of changes have come about.

The Bureau of the Budget has continued as the principal arm of the President within his executive office. There have been added, however, the Council of Economic Advisors, the National Security Council, and the National Security Resources Board, the Director of Defense Mobilization and, recently, the Office of the Director for Mutual Security.

Departmental Management

Management of federal departments and agencies has been strengthened in many ways during the decade. The most important over-all development has been the increasing realization that internal leadership, direction, and co-ordination of the day-to-day task of mobilizing men, materials, money into effective operation is often more important than questions of organization structure.

The Hoover Commission prepared no formal report on departmental management, but as a result of the pressures by the Bureau of the Budget, a staff report was prepared and published on this subject. This report, supplemented by a few recommendations in the Commission's over-all report on "General Management of the Executive Branch," has helped support the efforts of departmental executives to improve the internal organization and management of their agency.

The Commission recommended that the department and agency heads should be equipped with adequate staff assistants covering such fields as legal matters, finance, personnel, supply, management research, information, and Congressional liaison. The Commission also recommended that a department head should be given authority to determine the internal organization within his department.

The significant actions taken mainly through the submittal of reorganization plans by the President to Congress vested authority in the heads of agencies to manage their departments and empowered them to determine internal organization. This was achieved in five departments; only the reorganization plan for the Department of Agriculture was rejected by the Congress.

The President also has submitted a number of reorganization plans which vest in the chairmen of commissions responsibility for executive or internal administration. In most commissions this responsibility had previously been divided equally among all members. As a result even minor administrative actions might be impossible without the concurrence of a majority of the members. The reorganization plans for the Federal Trade Commission, the Securities and Exchange Commission, the National Security Resources Board, and the Civil Aeronautics Board were accepted; the Congress rejected such plans for the Interstate Commerce Commission, the Federal Communications Commission, and the National Labor Relations Board.

In a number of cases the reorganization plans provided for an Administrative Assistant Secretary. Such positions, in contrast to other assistant secretaries, are intended to be filled by appointment by the President without the political implication of confirmation by the Senate.

There has been an increased understanding in the Government of the importance of program planning, i.e., the translation of general objectives and purpose into concrete programs setting forth the activities, projects, and actions to be performed within time limits if the objective can be achieved. Although progress to date is promising, program planning and co-ordination in relation to basic objectives and goals continues to be one of the government's most perplexing problems. The evaluation of both goals and performance warrants further attention, also.

Another interesting development is the increased emphasis upon decentralization of authority to lower levels and especially to the field. Recently, the President required each agency to report the action that it had taken to delegate authority to subordinate levels and to eliminate the unnecessary checking and review at higher levels by officials who should be devoting their attention to more important problems.

The most significant development in federal management has been without question the greater attention given to the human factor in management. Federal executives, as those in private industry, have come to realize that high production is not the product of mechanics and gadgets; it is achieved when competent people work together under suitable conditions and under leadership which demonstrates its interest in them as human beings. Accordingly, federal agencies are giving increased attention to questions of selection and training of persons with appropriate aptitudes. Beyond this they are instituting supervisory processes which enable the worker to participate in decisions which are of importance to him and to feel that he is teamed up in an important endeavor to which he has a significant contribution to make. Federal experience is supporting the findings of Dr. Rensis Likert in his industrial studies that high productivity is the product of the employee-mindedness of supervisors and is not achieved by high-pressured emphasis on production goals.

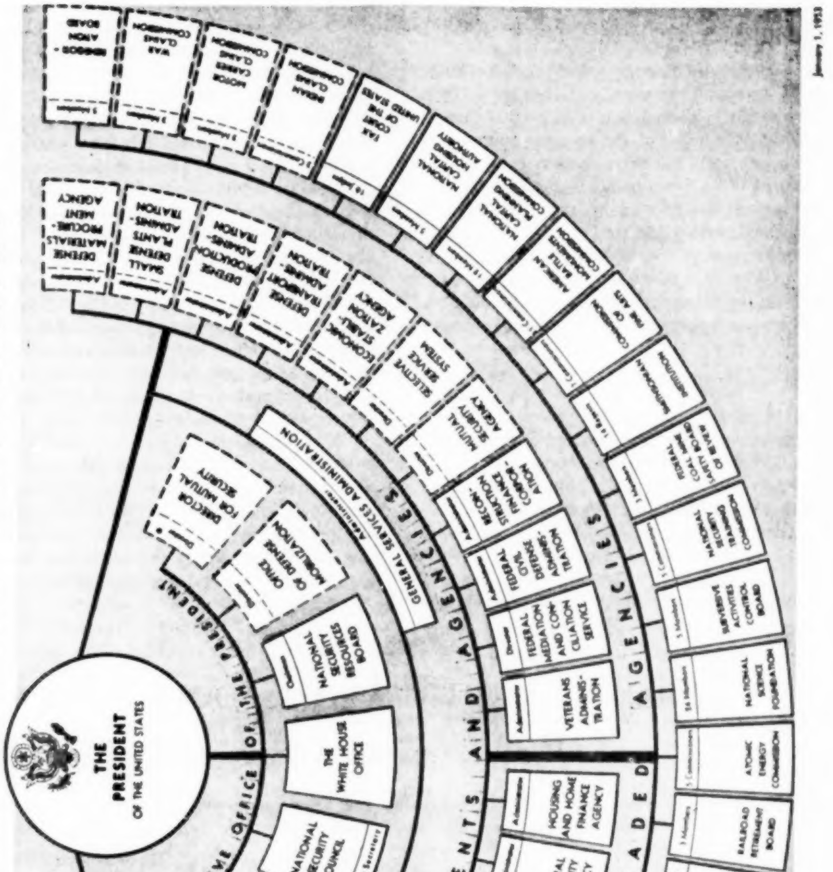
President's Management Problem

Finally, note should be made of the initiative taken by the President in establishing a government-wide management program. One part of that program encompasses the improvements in executive structure and departmental management described earlier in this article, many of which were made in furtherance of recommendations and principles proposed by the Hoover Commission. Another part deals with improvement of processes involved in government-wide personnel, fiscal, and property management. Still a third has stressed management improvement actions by agencies themselves through periodic review of their operations.

Major strides have been made during recent years in improving the management of government-wide processes. The vesting of administrative authority in the President of the Civil Service Commission, by a reorganization plan approved in 1949, was a first step toward strengthening the central administration and leadership of personnel management throughout the Government. The establishment, referred to earlier, of the General Services Administration as the government's central supply and service agency was another move that has strengthened government-wide activities.

Fiscal management has seen even more significant progress. The federal budget has emerged as a primary tool for the review and control of federal programs in relation both to the services provided by the Government and to the impact of federal expenditures and fiscal policies on the national economy. In addition, long-needed improvements in accounting and audit practice have been initiated. Of special importance has been the collaboration of the General Accounting Office, the Bureau of the Budget, and the Treasury Department in the development of a

EXECUTIVE BRANCH OF THE GOVERNMENT



joint program for improving accounting and audit functions and the enactment of the Budget and Accounting Procedures Act of 1950.

That part of the President's Management Program calling for periodic reviews of agency operations, initiated by Executive Order 10072, issued in July, 1949, has helped keep the agencies on their toes and made the heads of the agencies more management-minded. This program has been supported by Congress through the enactment of an amendment to the Federal Classification Act which, in addition to stressing the review of agency economy and efficiency, authorizes the payment of incentive awards to individuals or groups who make outstanding contributions to efficiency. The Bureau of the Budget gives general direction by reviewing the management programs formulated by executive agencies and assisting on problems of an interagency character.

Credits

The progress of management in the Federal Government throughout the decade could not have been reported in such an affirmative way had it not been for the pioneering work in governmental reforms initiated in 1906, by the New York Bureau of Municipal Research and for the new approaches to organization and management initiated under what has become the Scientific Management Movement by Frederick Taylor at the turn of the century.

The particular human resources which the emergency nature of the Government's problems called forth have had a telling effect on the management pattern. The general pattern set in 1932–1942 by the statesmanlike report of the President's Committee on Administrative Management continued to dominate central planning. But the influx, during the war and for special problems thereafter, of competent and dynamic individuals from private enterprise and from academic centers has had a marked impact on the total pattern of management.

We must credit also among the main forces affecting management in Government the increasing number of outstanding young men and women who desire to make public service a career; the increased attention which business and professional men have been giving to problems of Government; the development of professional associations of public officials in many fields; the rise of a great body of literature and knowledge regarding public management, the establishment of schools, foundations, and research institutions aimed at training persons for public service and in expanding the horizons of knowledge and practice of governmental administration.

Thus, during the decade, a number of different streams of management practices and thought have flowed into the Federal Government. To some degree these tributaries have retained their own characteristics; to some extent they have merged. At this stage it may be said that there is not a full assimilation of the varied approaches and influences, but the management resources of the Federal Government have been greatly enriched by the influx.

International Co-Operation

By HAROLD B. MAYNARD,[1] PITTSBURGH, PA.

The latter half of the past decade has seen a tremendous growth in the exchange of scientific-management information among most of the industrial countries of the world. During World War II, the United States had demonstrated to the world that it could produce far greater quantities of materials than anyone had believed possible before. Communications among countries were difficult, however, and the methods by which this production was achieved were little known outside our own country.

Consequently, when the war came to an end and people in the war-devastated countries were faced with the need for the immediate production in large quantities of the many things that were required to restore a normal way of living, they turned to the United States eagerly for information on how to increase productivity. This eagerness for information had its impact on individuals, on management societies, and on governments, and stimulated an interest in American management know-how that was unprecedented.

Barriers to Individual Activity

Normally, a good deal of the demand for management know-how would have been satisfied by individual initiative. Industrial firms in various countries, for example, have long had arrangements for obtaining patents, licenses of various types, management know-how, and the like, from one another. American management consultants have worked in countries around the world introducing American methods and techniques. After the war, however, the difficulties of making working arrangements of this type with American firms or consultants were greatly increased by currency-exchange barriers. It was difficult for Americans to receive payment for services in dollars, and since foreign currencies were spendable only in the countries in which they were issued, there was not much desire to accept payment in local money.

In spite of these difficulties, there has been a good deal of international co-operation on an individual basis. Such arrangements have been most effective and have made up in quality what they have lacked in quantity. When and if currency-exchange barriers are removed, an immediate upsurge in co-operative arrangements among individuals in various countries unquestionably will result.

Management-Society Activities

The management societies in various countries have been co-operating in the exchange of management know-how on a tremendously increased basis since the end of the war. As soon as hostilities ceased and communications were restored, the national management societies in various countries through CIOS, the International Committee on Scientific Management, began to plan for the Eighth International Management Congress. Because of the war, no Congress had been held since the Washington Congress in 1938. Therefore there was a great interest in Europe in organizing a Congress as quickly as possible and in securing American papers for it in all management areas to fill the gap in communications caused by the war.

The Eighth Congress was held in Stockholm in July, 1947. Over a hundred American delegates attended, for the most part traveling to and from the Congress in a group on the steamship *Drotningholm*. On the voyage over, the delegates held a series of preparatory meetings during which they discussed not only the management information in which it was felt the other countries

[1] President, Methods Engineering Council. Mem. ASME.

would be most interested but also how to present this information in a way which would be likely to gain its acceptance.

The Congress itself was ably organized by the Swedish National Committee. The papers and the sessions at which they were discussed provided the opportunity for the exchange of information on a formal basis. The social occasions—of which there were many, all excellently planned and executed—permitted the exchange of information on an informal basis. The informal exchanges were at least as valuable as the more formal sessions. They enabled individuals from different countries to become well acquainted and resulted in the formation of a number of international friendships. Many of these have endured ever since and have helped to keep the channels of international management communications open.

At the business sessions of the Congress it was agreed unanimously that CIOS had a vital function to perform in the postwar world in encouraging the free exchange of management information among its member national committees. It was voted to establish a permanent paid secretariat and to provide funds for operation which would permit CIOS to function between international congresses.

Growth of the National Management Council

The body representing the United States in international management affairs is the National Management Council. It was formed at the beginning of the preceding decade, namely, in 1933. Up to that time international co-operation in management affairs had been somewhat sporadic. It started off auspiciously enough in 1923 when the government of the newly created republic of Czechoslovakia under the leadership of its scientist-president Masaryk, invited a number of countries to participate in a meeting for the exchange of ideas and experience in the field of industrial management. At the Congress which was held in Prague in July, 1924, each meeting was presided over by two officers, one representing Czechoslovakia and one the United States. This Congress resulted in the creation of CIOS and the beginning of permanent international co-operation.

To direct American participation in other Congresses which were held in Brussels in 1925, in Rome in 1927, in Paris in 1929, and in Amsterdam in 1932, an "American Committee on Participation, International Management Congresses" was set up. It functioned on an informal, volunteer basis, without any substantial support from other management groups. Although it succeeded always in having American delegates and papers at the International Management Congresses, the representation from the United States at the Amsterdam Congress was so pitifully inadequate it became evident that some more effective means of fostering international co-operation was necessary.

As a result of subsequent discussion a joint invitation to consider what should be done was extended in December, 1932, to a number of American management groups by the presidents of the American Marketing Society, the National Office Management Association, the Society of Industrial Engineers, and the Taylor Society. A meeting was held on December 20, 1932, which was attended by representatives from these four societies and in addition representatives of the American Committee on Participation, International Management Congresses; American Management Association; The American Society of Mechanical Engineers; Association of Consulting Management Engineers; National Association of Cost Accountants; and Personal Research Federation.

This meeting was addressed by the late Dr. Harry Arthur Hopf. After tracing the development of the management movement both in America and abroad, Dr. Hopf concluded with the following recommendation:

"As a constructive proposal for joint consideration by the management associations and societies represented at this gathering, the president of the American Marketing Society, the president of the National Office Management Association, the president of the Society of Industrial Engineers, and the president of the Taylor Society have the honor to recommend to their colleagues in the management field the formation of a national management council and to invite their participation in such a co-operative undertaking.

"In advance of discussion of this proposal, it is hardly appropriate to refer to the practical aspects which will undoubtedly present themselves for consideration. The presidents of the four organizations named deem it fitting, however, to record the view that a national management council, in addition to addressing itself to matters of broad, general concern to the management movement as a whole, will be in a position to strengthen the relations of the constituent bodies to each other, to further their intensive development, and to provide a basis for group action whenever that may appear desirable."

The National Management Council was formed as a result of this meeting on June 8, 1933. It continued to sponsor United States participation in international congresses, culminating with the organizing of the Seventh International Management Congress which was held at Washington in September, 1938.

The war then intervened, and the National Management Council by mutual agreement remained largely inactive during the war years. When the war ended, it again came to life and organized the American participation in the Stockholm Congress in 1947.

Up to this point the National Management Council had always functioned largely without funds, relying on the voluntary services of interested individuals for getting its work accomplished. When at Stockholm, it was agreed that CIOS should operate on a full-time basis, it was evident that NMC also would have to be organized in a more permanent fashion. In addition, at Stockholm NMC had agreed to give CIOS substantial financial support, although it was itself without funds at the moment.

With new objectives and new commitments, NMC examined itself after the Stockholm Congress and revised its form of organization and its methods of operation. With the generous support of its management society members and with the aid of a loan—since repaid—from the societies, it set up and staffed a permanent office, adopted a new set of by-laws, and secured new members from management societies, commercial and industrial firms, and educational institutions. It co-operated fully with CIOS and with the national committees of other CIOS countries and began to develop channels and methods of international co-operation far more effectively than had been possible in the past. During 1947 to 1949, NMC managed to meet its commitments in full and to develop itself to the point where it was able to accept the unusual opportunity for international co-operation which presently developed.

Government-Sponsored Activities

The need for co-operative assistance to the war-ravaged countries of Europe which was recognized by the Stockholm Congress also was recognized by the governments of the countries concerned. The Anglo-American Council on Productivity was set up presently to facilitate the exchange of management and technical know-how between the United Kingdom and the United States of America. This was followed by the establishing of the Economic Cooperation Administration or the Marshall Plan with a Technical Assistance Division keenly interested in doing all that it could to encourage increased productivity. A little later the program of assistance to underdeveloped countries was proposed by the President of the United States in his famous "Point 4," and another area of co-operation in the field of international management was opened.

The government programs had behind them huge grants of money. They had the funds which the management societies lacked, but they lacked the management know-how which the members of the management societies had. This situation in the United States was quickly recognized by the National Management Council. Early in 1950 an arrangement was worked out between ECA and NMC, whereby the latter placed at the disposal of ECA its assistance in handling the teams of management people which ECA was beginning to bring from other countries in increasing numbers.

During 1950, 1951, and 1952, the National Management Council guided the training of a number of visiting teams. It did not limit its activities to this form of co-operation, however, but took the initiative in finding other ways of giving assistance. In 1951 it developed the mechanism of the management seminar. Under NMC leadership, small teams of American management experts were sent abroad to hold seminars on management subjects in various countries in Europe. The United States teams met with groups of foreign industrialists and technicians in their cities and discussed the application of management principles to their own problems. This seminar activity is continuing at the present time.

Another important event in the field of international co-operation took place in the fall of 1951. Realizing that an understanding of and a desire for increased productivity must permeate all levels of management before any appreciable results can be obtained, NMC recognized the necessity of bringing the story to top-management people in other countries. Working in co-operation with the National Association of Manufacturers and ECA, it organized what came to be known as "Operation Impact." This project brought over 250 top European executives to the United States in November and December, 1951. They were given the opportunity to meet with and talk freely to men of equal rank and with similar interests in this country. The result was that a new area of international co-operation and understanding was opened on a very important management level.

The co-operation between NMC and ECA not only assisted importantly in spreading a knowledge of sound management procedures throughout Western European countries, but it enabled NMC to earn the funds which it needed to meet its primary obligations. Thus it has been able to give increasing support to CIOS. Furthermore, these activities have not interfered with its other major purposes. Throughout 1949, 1950, and 1951, the National Management Council worked on preparations for the Ninth International Management Congress which was held at Brussels, Belgium, in July, 1951.

Ninth International Management Congress and Future Plans

This Congress developed an interesting new method of international co-operation. At previous Congresses the papers which were presented for discussion were prepared by individual authors. This of necessity somewhat limited the viewpoint from which they were presented. It was decided by the Executive Committee of CIOS that the papers for the Ninth Congress would be prepared by committees. The responsibility for the preparation of each paper was assigned to a specific country. This country set up a committee to take the lead in producing the paper. The other countries who were interested in contributing then set up collaborating committees. They worked with the main committee in the lead country, contributing reports which were included in the final report. Thus, in preparing the twelve papers which were presented at the Brussels Congress, over 600 individual management people co-operated.

It may be seen from what has been said that from a rather low prewar level and a period of practically no activity during the war, international co-operation in the field of management has been increasing at an accelerating rate since 1945. There is every reason to believe that it will continue to increase in the foreseeable future barring major dislocations such as war or economic disaster.

The International Management Congresses, for example, will continue to be held. The Tenth Congress is scheduled to be held in São Paulo, Brazil, in February, 1954. The papers will be prepared again by the method of international co-operation just described. In addition, it is planned to subdivide the congress delegates into small groups of 25 to 50 so that they will be able to discuss the papers in small, intimate, round-table meetings. This not only will provide the opportunity for freer discussion, but it will enable the delegates to become better acquainted.

The 1957 Congress has already been spoken for by both Australia and Switzerland. Although it probably will be more convenient for the Europeans to hold it in Switzerland, it is interesting to observe the desire of the Australians to play an increasingly active part in international management affairs.

At the moment of writing there are 16 countries who are members of CIOS. Several additional applications for membership are pending. Through the activities of CIOS, the member national committees are being encouraged both to greater efforts to spread scientific management within their own countries and to fuller co-operation with the management committees in other countries.

The stimulus to international co-operation provided by government will continue in the future. The Technical Assistance Division of the Mutual Security Agency—the successor to ECA—is currently greatly interested in stepping up education for productivity in many countries. Its Advisory Group on European Productivity through its Subcommittee on Education for Productivity is making an informal survey of the problems involved which will lead to specific recommendations. This survey is being conducted with the assistance of over a hundred management people in other countries—another evidence of the way international co-operation is increasing.

It is apparent that we are now living in an international world. Problems which formerly were considered as local or national in nature are now being discussed on an international level. Management is no exception. People everywhere in the world are increasingly insisting upon a better way of life. They are demanding more in the way of material things. To be able to have those things it is necessary to increase productivity everywhere. This is obvious not only to the managers of industry but to governments and workers and all people everywhere. Interest in increasing productivity is on the ascendent, and people are looking to America to help them find the answers. American management—although humbly aware that it does not know all that there is to know and anxious to make the exchange of management information a two-way process—has shown a statesmanlike recognition of its obligation to help others less fortunate in every way that it can. Since 1945, international co-operation has been more than a mere phrase. It has become a practical reality. It will continue to be practiced on a widening scale in the decade which lies ahead.

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